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THE EXPERIMENTS MADE BY MR. UTHEMANN TO DISCOVER A PROCESS FOR PREVENTING THE CORROSION OF COPPER AND BRASS BY SEA-WATER UNDER THE CONDITIONS FOUND IN THE SURFACE-CONDENSERS OF MARINE STEAM-ENGINES.

Translated from "Le Génie Civil" of the 23d of September, 1905,

BY CHIEF ENGINEER B. F. ISHERWOOD, U. S. NAVY, MEMBER.

Mr. Uthemann, a Member of the Engineer Council of the German Navy, has recently made a series of experiments to discover a process that would prevent the corrosion of copper and of its alloys by sea-water. Before giving the results of his experiments, some facts relative to this problem may be recalled as germane to it.

The employment of copper-sheathing for the immersed exterior part of the hulls of wooden sea-going vessels dates back to 1761, when the British Admiralty, for the first time, commenced its use for preventing the adherence of barnacles and similar accretions to their hulls.

After the introduction of steam machinery into sea-going vessels, copper and its alloys soon superseded cast iron for the enormous quantity of piping required and for other purposes, on account of its less weight and bulk, properties of great value in that case, and which became more and more important as the power to be developed in the same sized vessels became greater and greater.

In proportion to the greatness of this development, the corrosion of copper, which was so little at the beginning of its use that it might be considered negligible, became important. Thus, in 1883, this corrosion was from five to six times as much as previously. Stolzel attributes that increase to the increase of the impurities in the copper The impurities in question were derived from the mineral matter in the foreign ore imported into England to meet the increased demand for the metal, and were in a very much greater proportion than in the purer native ore that had previously furnished the copper needed.

At this epoch Davy experimented on the protection of the sheathing-copper of wooden sea-going vessels by placing it in contact with electro-positive metals that were more easily attackable, such as iron and zinc. He ascertained that the protection was complete as long as the surface of these metals was maintained at from the $\frac{1}{40}$th to the $\frac{1}{150}$th of the surface of the copper to be protected. But, unfortunately, this process produced carbonates of lime and of magnesia which increased the development of barnacles and of algae, this causing a still greater increase in the resistance of the vessels. Nevertheless, further researches, confirmed by practice, showed that by adopting from the $\frac{1}{150}$th to the $\frac{1}{1000}$th for the ratio of the surfaces referred to, this disadvantage disappeared and the copper remained clean and free from attack. Ultimately, however, the method was definitively abandoned owing to practical reasons. Later, the investigation was again undertaken in the laboratory of the Toulon arsenal, by Becquerel, who attributed the practical failure of the method to the fact that the protecting metal disappeared much more rapidly than had originally been foreseen.

Since 1883, this question has again arisen, not for the protection of the external sheathing of wooden vessels, but for the preservation of the copper and brass tubes of the surface-condensers of marine steam-engines, and for the preservation of their brass feed-pumps and bilge-pumps, etc., the rapidity of the destruction of which in a large number of merchant ocean-steamers seemed so great as to have caused the abandonment of copper and its alloys as too costly in favor of the use of wrought and of cast iron, notwithstanding the inconvenience of the frequent renewals of the latter needed. The cause of this corrosion of copper and of its alloys, is certainly of an electrolytic nature, and should be attributed to the impurities in the metal, but the kind of these impurities and the forms under which they are present play a more important part than their proportion in the metal. The remark should be here made that copper refined in the dry way by the ordinary processes, but still containing a little oxide, is less attackable than the chemically pure copper obtained by electrolysis.

The alloys of copper have given no more satisfactory results than that metal itself. The demand of the specifications, for example, that the foreign matters should not exceed six-tenths of one per centum of the copper, has no great importance as regards resistance to the attack of sea-water.

Finally, the concentration, the composition, and the temperature, of the sea-water, seem to have great influence on the rapidity of the corrosion. Thus, a sea-water more concentrated with marine salt, richer in chloride of magnesium, and hotter, as in the case of the condensing water flowing from a surface condenser; or, simply in the case of tropical sea-water near certain coasts, is greatly the most corrosive.

All these facts show clearly that no great protection is to be had from any modification in the composition of the copper, and that the best chance is an investigation whether a coating or covering—metallic or non-metallic—could be found that would hermetically separate the sea-water from the metal; such, for instance, as tar, asphalt, caoutchouc, varnishes, lac-

quers, tin, zinc, and the American coating invented by Sabine, which is one of the best. Nevertheless, none of these coatings has given satisfactory results. All of them are too fragile, wear off too rapidly and scale off; further, most of them very easily crack. The presence of these cracks causes the metal exposed by them to be attacked with much more intensity than if it had no protection.

Lead should make a sufficiently good covering, but, unfortunately for its efficiency, its thickness would have to be at least 0.12 inch, which would increase the weight of the tubes, diminish their diameter, augment their resistance to the flow of the condensing-water, and, finally, prevent the entrance of the expanding tool in those of small diameter.

Mr. Uthemann has likewise had recourse to a covering, but of a different nature from the preceding ones; it is an oxide of iron deposited by a strong electrolytic action upon the copper immersed in a mass of sea-water. The idea of such a coating was suggested to him first by the experiments of Davy and of Becquerel, and then by the well known practical fact that the copper or brass tubes of surface condensers with cast iron shells are but slowly attacked by sea-water.

As an intense electrolytic action can be produced at only a short distance, the piece of copper or brass to be protected is enveloped by a wire or band of iron preferably to a band of zinc for the material, wound spirally around the piece; the zinc is, in fact, less electropositive than the iron, and, further, it has the inconvenience of dissolving in sea-water. If a copper-tube thus prepared be immersed in sea-water there will be a copious disengagement of gaseous bubbles and the liquid becomes simultaneously of a reddish-yellow color by reason of the formation of the oxide of iron at the expense of the iron.

At first, the copper remains bright, then the oxydation of the iron follows and the copper becomes covered between the spirals with iron which soon becomes sufficiently adherent not to be swept off by a very rapid current of water, even after the removal of the iron spirals. The same deposit is pro-

duced, but more slowly, under the spirals themselves. If the coating be scratched by a scraper or sharp edged tool, the copper beneath appears bright and unchanged. When all the surface of the copper is covered with a sufficiently thick coating, the attack of the iron seems to cease and the piece can be removed. The copper can then remain indefinitely in the sea-water without alteration, provided the coating be maintained in its integrity. The same results are obtained with the alloys of copper.

If the tubes, or other pieces, be formed of pieces of different metals soldered together, one piece being of copper and the other pieces being of brass or other alloys of copper, the corrosive action is the same. Coated tubes, that is to say, tubes coated with tin or zinc, resist corrosion better than non-coated tubes.

An experiment made with a copper tube coated with iron electrolytically deposited upon it showed that this coating of metallic iron dissolved entirely off in sea-water, and affords no protection because the surface of the copper becomes rapidly exposed. Only the oxide of iron produced by the process hereinbefore described is protective, and not the metallic iron itself. As the protection is the same whether given to copper or to its alloys, the employment of pure copper is preferable to the employment of the alloy of 97 per centum of copper and 3 per centum of zinc which has frequently been recommended as less attackable by sea-water, but it is more expensive and not as easy to work as copper.

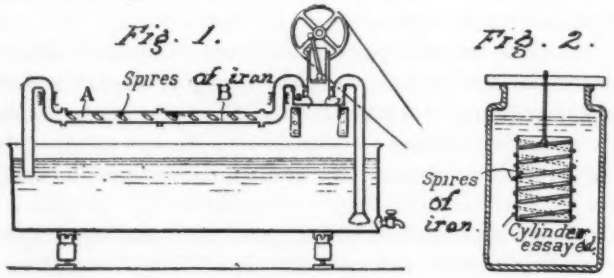
This mode of protection is just as properly applicable to the piping of bilge-pumps, as the water pumped by them is generally acid and strongly corrosive; and to the channels of frigorific apparatus using a highly concentrated brine as the non-congealable liquid.

Mr. Uthemann has made a great number of experiments of the kind precedently described. He has operated on copper tubes subjected to both flowing water and to stagnant water. These experiments have extended over two years.

The arrangements adopted in the first case are represented

in Figure 1. A and B are the tubes operated on; they are placed in the circuit of an electrically driven pump which draws the water from a reservoir situated beneath them and delivers it into the same reservoir after its passage through the tubes.

The conditions under which these experiments were made, reproduce as closely as possible those of the copper-conduits



ARRANGEMENTS ADOPTED FOR THE EXPERIMENTS WITH FLOWING WATER AND WITH STAGNANT WATER.

of sea-going vessels. The tubes A and B, like them, are subjected to the double influence of air and of sea-water which are the most favorable to their destruction.

The apparatus was in use ten hours per day, and, owing to the arrangement of the tubes, the water remained in them during the rest of the day when the pump was not working.

The water was renewed every fortnight, and the temperature of the laboratory was maintained at 68 degrees Fahrenheit.

The accompanying table contains the results of these experiments.

Tubes of electrolytic copper and of ordinary copper were experimented with. These tubes were cleaned, polished and weighed, before each experiment. The experiments were simultaneously made, and under precisely the same conditions, on protected and on non-protected tubes, consequently, the results are rigorously comparable.

Also, the composition of the sea-water and the mode of assembling the tubes were varied; and, finally, for ascertaining whether the protection were localized or communicated to adjacent parts, an experiment was made in which only one of the two tubes was protected. After each experiment, the iron wire wound spirally around the tubes, which constituted the protection, was taken off, the tube cleaned and washed with water, alcohol, and benzine, and then weighed.

Any increase of weight (due to the deposition of the oxide of iron) indicated protection; any decrease of weight indicated lack of protection, and, therefore, an attack on the copper; this loss of weight increases with increase of time.

The experiments in stagnant water were made on cylinders of thin copper, arranged as shown in the marginal figure 2; that is to say, in a glass vase. Cylinders were tried both of brass and of copper, or made one-half of each of these metals.

The tubes, both of copper and of brass, used in surface-condensers, were also experimented with by this method. All these tubes were, like the others, polished and weighed before being experimented with. The spirals of protection were applied either solely on the interior, or, simultaneously, both on the interior and the exterior of the tubes. The temperature was 68 Fahrenheit degrees, and the sea-water, taken from the Gulf of Danzig, was vaporised until it contained $3\frac{1}{2}$ per centum of salts. Mr. Uthemann employed a solution of 3 per centum of ordinary salt and 0.5 per centum of chloride of magnesium. Finally, for the purpose of ascertaining whether the protection of the oxide of iron was obtained in acidulated water, he added, also, in certain cases, 0.1 per centum of hydrochloric acid.

At the commencement of the attack of the iron, there was formed on the protector a great quantity of rust which necessitates rinsings of the tube before its use. This formation of rust deteriorates but little the protecting steel spirals; one of these spirals one-fiftieth of an inch in diameter, after a year's use had lost nothing of its cohesive strength or of its elasticity.

TABLE CONTAINING THE RESULTS OF THE EXPERIMENTS MADE BY MR. UTHEMANN, MEMBER OF ENGINEER COUNCIL OF THE GERMAN NAVY, TO DISCOVER A PROCESS WHICH WILL PREVENT THE CORROSION BY SEA-WATER OF COPPER AND BRASS UNDER THE CONDITIONS FOUND IN THE SURFACE CONDENSERS OF MARINE STEAM-ENGINES.
"Le Génie Civil" of 23d September, 1905.

Number of the Experiment.	Duration of the Experiments, in hours.	Metal essayed.	With or without protection.	Increase or diminution of weight, in fractions of a pound per square foot of surface of the metal essayed	Conditions.
				1. In Flowing Water.	
1a	94	Cast copper	With	+0.017026322	Water containing 3.5 per centum of marine salts. Temperature 55.4 degrees Fahrenheit.
1b	94	do.	Without	+0.001980663	
2a	52	Cast copper	With	+0.020224908	Water containing 3.5 per centum of marine salts. Temperature 60.8 degrees Fahrenheit.
2b	52	do.	Without	+0.007922653	
3a	30	Electrolytic copper	Without	−0.005240751	Water containing 3.5 per centum of marine salts. Temperature 60.8 degrees Fahrenheit.
3b	30	do.	Without	−0.008527924	
4a	187	Electrolytic copper	With	+0.034807621	Water containing 3 per centum of marine salts, and 0.5 per centum of chloride of magnesium. Temperature 64.4 degrees Fahrenheit.
4b	187	do.	With	+0.041596387	
5a	187	Electrolytic copper	Without	−0.025721556	Water containing 3 per centum of marine salts, and 0.5 per centum of chloride of magnesium. Temperature 64.4 degrees Fahrenheit.
5b	187	do.	Without	−0.036089897	
				2. In Stagnant Water.	
1a	59	Electrolytic copper	With	−0.021715942	Sea-water containing 3.5 per centum of marine salts, and 0.8 per centum of hydrochloric acid. Temperature from +41 to −4 degrees Fahrenheit. Iron spirals on the outside only.
1b	59	Brass	With	−0.032163017	
1c	59	Commercial copper	Without	−0.180326464	
2a	33	Electrolytic copper	With	+0.003533208	Sea-water containing 3.5 per centum of marine salts, and 0.8 per centum of hydrochloric acid. Temperature from +41 to −4 degrees Fahrenheit. Iron spirals on the outside only.
2b	33	Brass	With	+0.006677664	
2c	33	Commercial copper	Without	−0.004345157	
3a	198	Electrolytic copper	With	+0.013163414	Sea-water containing 3.5 per centum of marine salts. Temperature 64.4 degrees Fahrenheit. Iron spirals on the outside only. The cylinders were covered before commencing the Experiment with a deposit of electrolytic iron.
3b	198	do.	Without	−0.027232273	
3c	198	Brass	With	+0.005181710	
3d	198	Commercial copper	Without	−0.212829023	
4a	74	Cast copper	With	+0.001001404	Sea-water containing 3.5 per centum of marine salts. Temperature 71.6 degrees Fahrenheit. Iron spirals on the outside only.
4b	74	do.	Without	−0.027360217	
4c	74	Brass	With	+0.000058559	
4d	74	Commercial copper	Without	−0.020766208	
5a	74	Condenser tube of Heckman bronze	With	+0.001933915	Sea-water containing 3.5 per centum of marine salts. Temperature 71.6 degrees Fahrenheit. Iron spirals on both the outside and the inside.
5b	74	Brass	With	+0.000880841	
5c	74	Heckman Bronze	Without	−0.024456884	
5d	74	Brass	Without	−0.014762707	

NOTE BY THE TRANSLATOR.

The process set forth in the foregoing translation, is of a wholly new and very interesting method of preventing the corrosion by sea-water of copper, brass, and bronze under the conditions as nearly as they could be reproduced in which these metals are employed for the surface liquefaction of the exhaust-steam in the condensers of marine steam-engines. As all the other processes which have been tried for this purpose have been failures more or less signal, an entirely new one, supported by extensive experimental results, has great interest in steam-engineering, and even if it should not prove to be practicable, itself, on a commercial scale, it may be valuable as a stepping-stone to something that will be commercially successful in a very important engineering subject. The process in question, and its results under the experimental conditions, of course, cannot be accepted as a finality until tested for a considerable length of time on the large scale and under the conditions of actual practice. But, under any circumstances, the facts developed are suggestive to all physicists and increase largely the sum of this kind of knowledge.

The general facts regarding the corrosion by sea-water of copper and brass, are substantially the same in the case of the sheathing by these metals of wooden sea-going vessels, and in the case of the copper and brass tubes employed in the surface-condensers of marine steam-engines.

In both cases, the corrosion in question is of two kinds, one a uniform oxidation of the metal without sensible electrical phenomena, commencing with the first exposure and continuing until the final disappearance of the metal, the oxide and other products chemically produced having scarcely any adherence to the metallic base on which they are formed, and on which they merely rest as a sort of dust that continually washes off when the metal is laved by water, the washing off being so much the more rapid as the velocity of the washing water is greater. This continual oxidation and washing off of the oxide, constitutes the peculiar fitness of copper and brass sheathing for sea-going wooden vessels as a preventive

of the accumulation of marine animals on their immersed surfaces, a growth that not only greatly increases their resistance to movement in water, but, in the seas infested with the "teredo-navalis," prevents the destruction of the wood itself by these ship-worms. In the case of the metallic sheathing, however, the continual washing off of the non-adherent oxide carries with it this sea-life as fast as formed upon it, and leaves the metal clean, thus avoiding the increased resistance due to such an accumulation. The sea-life develops as rapidly upon the oxide as it would upon the unprotected wood. There is no poisonous effect produced on the sea-life by the metal or by its oxide, the barnacles simply fall off as the non-adherent oxide to which they clung falls off. Neither does the smoothness of the metallic sheathing prevent the attachment of the sea-life to it, as that attachment takes place to glass as rapidly and as firmly as to wood. The entire beneficial effect is due merely to the fact of the non-adhesion of the oxide of copper and brass to its metallic base, being in this particular the very opposite of the strongly adherent oxide of iron to its metallic base.

Sea-water is a solution of strongly alkaline salts, and its corroding effect upon copper and brass is in some ratio to the percentage of these salts in it. This effect is increased by increased temperature of the solution and by the rapidity of the movement of the metals in it, and is decreased in some ratio to the mass of the metal exposed, all other things remaining the same. As the mass of the exposed metal is increased, not only does the *percentage* of the weight of it corroded in equal time decrease, but the *absolute* weight of it corroded in equal time decreases also. This is an effect due wholly to mass, the greater mass having greater resisting power to chemical reaction. A neglect of this fact in many experiments on this subject, have caused erroneous *quantitative* conclusions, although the results may hold true *qualitatively*. The specimens of different metals experimented with should have the same weight, and, as nearly as this premiss will allow, the same dimensions.

The copper and brass sheathing of wooden vessels corrodes faster in sea-water when the vessels are in motion than when they are motionless, because, in the latter case, there is none of the washing action that exists in the former case and removes the oxide as fast as it is formed. When the vessels are motionless, the oxide adheres longer to its metallic base, and thus, in measure, reduces the rapidity of the corrosion by acting as a covering to the metals. The same result due to the same cause occurs with the copper and brass tubes of the surface condensers of marine steam-engines. But here must be remarked the important fact that as vessels not under way generally lie at the wharves of cities, and that as the sea-water in which they float is always more or less much polluted by the sewage discharged into it, the rapidity of the corrosion is increased, and often in a high degree by this pollution, so that in comparing the amount of corrosion produced in equal time with the vessel in motion and at rest, allowance must be made for the fact that the sea-water in the two cases is generally not of the same composition. Further, even were the sea-water of the same composition, there still remains the disturbing fact that the vessel at the wharf is subject to the movement of the tides and other currents over its sheathed surface which prevent a strictly quantitative comparison of the corrosion results in the two cases of motion and at rest. In this respect the conditions for the sheathing varies from those for the tubes in the condenser; the corrosive solution does not move over them when the vessel is at rest, and ought not to be present, but, as owing to the leakage of sea-valves, the condenser in such case is generally filled with outside water up to the outside level, and which remains in them practically motionless, the corrosion of the tubes will be less when the vessel is at the wharf than when it is in free route, the composition of the sea-water being the same, nevertheless, corrosion even then will be more or less in progress.

The corrosion hereinbefore described, is, as was stated, general in character and uniform in effect; it was a coating removed from the entire surface of the metal, leaving the

remainder clean and smooth. This corrosion was comparatively slow in its progress, and the injury done by it was not commercially important. Of course, it involved, after a time, a replacement of the metal, but the time at which this had to be done could be long foreseen, so that the inconvenience was comparatively small.

But there is another kind of corrosion in progress nearly simultaneous with the corrosion just described, and it is greatly more destructive and inconvenient. It is local instead of general, and attacks the metal in the most capricious manner, leaving some portions untouched while adjacent portions are absolutely destroyed to dust and gone. This is the kind of obvious corrosion which is attributed to want of homogeneity in the metal, and to electrical action, when the metal is in contact with sea-water. It commences by the appearance of extremely small holes—pinholes—more or less near each other, but only in particular spots on the metal, which holes gradually enlarge until they become sometimes confluent like the pustules of the smallpox, and sometimes they permanently preserve their circular form, the corrosion ceasing at that place and not extending farther, the edges or sides of these holes being nearly at right angles to the surface of the metal, and showing nearly the original thickness of the metal. They appear like holes artificially bored through. In this manner half of a tube in a surface-condenser will sometimes be completely destroyed while the remaining half will show no appearance of attack. Again, this kind of corrosion will continue slowly until it extends throughout the entire tube, which, while preserving its general form, has lost weight, color, and cohesive strength, and crumbles into dust by handling. Given, however, sufficient time, the copper or brass tube will disappear, having been apparently wholly dissolved by the sea-water. Or, more properly, the metal having become wholly chemically combined with the marine salts, and with the gases dissolved in the water, the results of the combination being various products in the liquid state, none of the metal being recoverable, as such, by evaporation. The pro-

cesses of the various reactions are exceedingly complex and obscure, and have never been convincingly investigated. All that is known is the final effect as shown by practice on the large scale and extending over, comparatively, considerable periods of time. Laboratory experiments are indeed most valuable, but only as guides to trials under the actual conditions of practice. As regards either palliatives for or preventions of, the corrosion in question, the commercial success involves not only the chemical success but also the money cost of the system; if that cost be greater than the cost of replacing the deteriorated metal, the method will be practically a failure however great its scientific success.

The same processes, appearances, and results, characterize the corrosion of the metal when employed as an external sheathing to the immersed part of the hull of a sea-going wooden vessel, as in the case of the tubes of the surface-condenser of a marine steam-engine.

What has hereinbefore been stated, refers to the corrosive action of pure or normal sea-water, but when this water is mixed with putrescent vegetable matter, reactions take place between them which convert the sea-water into what is known as "bilge-water," the corrosive power of which on copper and brass is much greater than that of normal sea-water, so that the destructive action of the latter upon these metals is very considerably accelerated.

The term "bilge-water" shows that this water was first observed in the "bilges" of wooden sea-going vessels, the putrescent vegetable matter required to be mixed with normal sea-water to form it, being supplied in that case by the decaying wood of the vessel, while the sea-water was supplied by inleakage from without. Bilge-water will rapidly destroy copper, brass, and iron pipes, made of *thin* rolled metal exposed to it; but cast copper, cast brass, and cast iron pipes, being made much *thicker* in material, resist much longer under the same circumstances than in the ratio of the difference of the thicknesses of the rolled and cast material.

Again, bilge-water is found on a large scale at the wharves

of cities laved by ocean tides, and whether these wharves be built of wood or stone; the putrefying vegetable matter being abundantly furnished by the sewerage discharge of these cities into the sea-water at their wharves. Hence, the rapid destruction that has been noticed in many cases of the copper and brass sheathing of wooden vessels lying for any considerable time at such wharves. Moreover, the iron boilers of steam-engines fed with this water are found to be not only rapidly corroded out, but kept free of sea-water scale by using this wharf-water; and the same effect is produced by feeding the boilers of marine steam-engines having jet condensers on board of wooden sea-going vessels with bilge-water instead of with normal sea-water from the hot-well.

Bilge-water is the best of all removers of sea-water scale from boilers, but it will, at the same time, attack their material.

In the chemical reactions between putrescent vegetable matter and the various dissolved salts of normal sea-water, the sulphur salts of the latter appear to furnish by their decomposition the sulphur which combines anew with the decaying woody matter. These new sulphur-compounds seem to be the corrodents carried by the bilge-water as a vehicle, and they are as corrosive in their vaporous as in their liquid state. They vaporise with the water, and have the strong and nauseating stink which characterises bilge-water; so much sulphur, so much stink. This vapor also reacts energetically on silver, on iron, on white-lead paint, and on a great many other substances.

The fetid odor of the gases continuously discharged from bilge-water compared with the nearly odorless steam evaporated from normal sea-water, and the strongly corrosive reaction of those gases compared with the neutral effect of steam from normal sea-water, show how great are the chemical transformations produced by the addition of putrid vegetable matter to sea-water. The transformations in question occur at all temperatures above the freezing point of the water, but they take place the more rapidly the higher the temperature.

Healthy wood, without rot and entirely submerged in sea-water, does not undergo decay as long as it is kept submerged. In that condition it will remain unaffected indefinitely, but it must not be alternately submerged and exposed to the atmosphere at intervals; the immunity only exists under the condition of uninterrupted exclusion of the atmospheric gases; as soon as they obtain access to it decay commences.

The *Chicora*, a wooden sailing merchant vessel employed in the China trade, was newly sheathed in January, 1847, with 7,392 pounds of copper, which was corroded so rapidly that the vessel had to be resheathed in March, 1849, only 2,682 pounds of the metal remaining. Time, 26 months.

The *Hamilton*, also a sailing wooden merchant vessel employed in the India trade, was newly sheathed in October, 1847, with 7,706 pounds of annealed copper, which corroded so rapidly that it had to be removed in August, 1849, only 3,086 pounds of the metal remaining. Time, 22 months.

The *Carthage*, a wooden sailing merchant vessel, also employed in the India trade, was newly sheathed in November, 1847, with 8,727 pounds of cold-rolled copper, which was so rapidly corroded that only 5,810 pounds remained when the vessel was resheathed in August, 1849. Time, 21 months.

The average duration of copper sheathing on American wooden vessels engaged in foreign trade, was 3 years, at the expiration of which time what remained of it had to be removed. The sheathing was of ordinary commercial copper.

The wooden vessels of the United States Navy, were formerly sheathed with copper refined to the highest degree in the Washington Navy Yard where the Government maintained a special plant with which no expense was spared in purifying the sheet-copper. The average duration of this sheathing on these vessels was about 5 years.

A number of years ago a wooden vessel belonging to the British Navy, and engaged in the suppression of the Slave Trade on the western coast of tropical Africa, had its copper-sheathing destroyed in an unprecedentedly short time. This vessel lay closely off a low coast of swamps of immense extent,

the tide covering and uncovering them alternately; the decaying matter of the swamp vegetation becoming largely mixed with the surface sea-water, formed doubtless the resulting principal corrosive agent, namely, sulphuretted hydrogen.

Similarly, the land above the Mare Island Navy Yard in California, is covered to an enormous extent with fresh-water swamps, the water of which, impregnated with the putrid parts of this continually dying vegetation, flows slowly past the wharves of the Navy Yard where it is mingled with the mixture of sea-water and fresh water from the adjacent bay and the rivers emptying into the bay. This mixture has an average density of five-eighths of the density of natural sea-water, and is so corrosive that the iron boilers of the shops of the Navy Yard using it for feed never form scale; their inner surfaces are always clean and undergo rapid destruction by corrosion.

The feed-water pipes and the bilge-water pipes of the United States wooden steam-frigate *San Jacinto* were wholly destroyed by corrosion in about 5 years. These pipes were of copper, and being situated in the bilge of the vessel were exposed to the action of the bilge-water which was the corroding agent.

During the same time (about 5 years), the tinned brass tubes of the Pirsson surface condenser of the engines of the *San Jacinto* were so much corroded by the sea-water passed over them for the liquefaction of the exhaust steam that they had to be removed. The shell of the condenser was of cast-iron, and but slightly affected.

The United States screw-steamer *Princeton* was constructed of green white oak. The bilge-water in this vessel was extremely fetid, so much so that the engine-room personnel had to hold their noses when they first went on watch until they became gradually accustomed to the stench. When the vessel was under steam there was a copious admission of sea-water to the bilges at the stern stuffing-box of the screw-shaft. The engines, two in number, had jet-condensers, and the boilers were fed with sea-water from their hot-wells. The *Prince-*

ton had no bilge-pumps worked by the engines, but the bilge-water was pumped out by the vertical single-acting air-pump of one of them, which pump was fitted with a snifting-valve at its bottom; whenever this valve was submerged by the bilge-water, it was opened by hand and held open until the bilge was cleared of water. Now the bilge-water, by this arrangement, was pumped into the hot-well and thence overboard through the outward delivery-valve, a part of it passing from the hot-well with the normal hot-well water into the boilers as a portion of their feed. The bilge was pumped out about every two hours. The density of the water in the boilers was maintained at twice the density of sea-water, and the steam-pressure averaged about 12 pounds per square inch above the atmosphere. Yet no scale ever formed on the interior surfaces of the boilers, but their metal corroded with extraordinary rapidity. In about 4 years' use they were corroded beyond repair, although, during that time, they were once extensively repaired. In about the same time (4 years' use) the brass of the air-pump, the feed-pump, and of the outboard delivery valve, and the cast-iron of which the connecting passages was made, had become so corroded as to need renewal, while the copper feed-pipe, which was situated low down in the bilge, was continually under repair. Now the corresponding parts of the other engine, which had been subjected to the action of only the normal sea-water of its air-pump discharge, remained unaffected, showing that the corrosion in question was due wholly to the bilge-water pumped overboard by the engine discharging it.

The corrosive effect of sea-water on metal is much intensified by mixing it, or supersaturating it, with atmospheric oxygen as in the case of spray. A striking example of this was shown at Vera Cruz in Mexico during our war with that country. Two medium sized cast-iron cannon had been placed vertically at the foot of the Mole or long pier extending into the sea; they were sunk, to act as posts, into the earth for about one-third of their length, and during strong winds were exposed to the dash of fine spray driven against

them from the sea, and always in the same direction; the result was that the side of the cannon receiving the spray had been so much corroded that about one-third of the material had disappeared, the cannon having a flat surface on that side instead of a round surface, while the opposite side was not affected. The cannon were about 50 feet apart, and both were corroded in the same manner and to the same extent. It is quite possible that the sea-water pumped as injection through the tubes of the surface-condenser of a marine steam-engine, and supersaturated with atmospheric oxygen due to the dashing of the waves into the air, caused by movement of the vessel, or by winds, may, in this manner have its corrosive power considerably increased. In other words, the corrosive effect of sea-water supersaturated with oxygen, is probably much greater than when not thus supersaturated.

There is still in the case of the surface-condensers of marine steam-engines, another possible cause increasing the corrosive effect of normal sea-water, namely, the electricity developed by the friction of this water upon itself and upon the metallic surfaces on which it flows.

However small these causes may seem to be, *per se*, yet when the relatively trifling weight of the metal subjected to the corrosion is considered in comparison with the very large weight of water coming in contact with it during considerable intervals of time, even a small increase of the corrosive power of the water becomes important commercially.

Some years ago, a large United States wooden steamship was fitted out at the Brooklyn Navy Yard for an extended cruise. The vessel which had just been newly coppered, lay in tide water at the Navy Yard Dock, and opposite to the discharge of one of the large sewers of the City of Brooklyn. The result was that when the vessel arrived in direct transit at Rio de Janiero in Brazil, the copper sheathing was so completely corroded that it had to be entirely renewed; it was probably greatly disintegrated before the vessel left the dock. The mixture of the putrefying vegetable matter of the sewage with the sea-water had formed a liquid (real bilge-water) that

dissolved the copper. The stench of the sulphuretted-hydrogen gas emitted by this liquid was nauseating in the extreme. The dock was a wooden platform resting on wooden piles partly out of water and partly in water. The tidal sea-water flowed among and around these piles.

The following are the generalisations which may be considered as warranted by the observations made on the large scale under the conditions of practice, regarding the corrosion of copper, brass, and bronze, in normal sea-water, and in sea-water contaminated by putrefying vegetable matter (bilge-water).

(*a.*) Normal sea-water contains dissolved in it sulphur compounds. Putrefying vegetable matter is dead vegetable matter, the constituents of which are continuously undergoing changed relations of chemical affinity, the affinities that existed between them in the living vegetable organism becoming continuously lessened to complete extinction in the dead vegatable matter, so that new chemical combinations can easily take place. Especially in this the case as regards the hydrogen-compounds is vegetable matter; the force of the chemical affinities of the constituents of these compounds greatly lessens in the change of the vegetable matter from life to death.

When, therefore, the dissolved sulphur-compounds in the sea-water, are presented to the hydrogen-compounds in the changing dead vegetable matter, there occurs a double decomposition, and a new compound, sulphuretted hydrogen, is formed; sometimes, also, there can be formed, additionally, hydrochloric acid by the combination of the hydrogen freed in the nascent state in the decaying vegetable matter with the chlorine of the dissolved common salt in the sea-water. Now, both the sulphuretted hydrogen and the hydrochloric acid have excessive corrosive action on metals. Thus may be easily seen that, according to circumstances, according to the more or less decay of the vegetable matter, the corrodents so generated in equal time in a given mass of sea-water, may be greater or less in quantity and in virulence of action;

and that the corrosive action due to the alkaline salts dissolved in normal sea-water may be greatly added to by the new sulphur compound, and by other compounds, generated as above described, and which convert normal sea-water into bilge-water, the latter being by no means confined to the bilges of wooden ships, but being found wherever sea-water is mingled with putrefying vegetable matter.

For normal sea-water, the corrosive action is due purely to the alkaline salts dissolved in it and which undergo decomposition in the presence of the metal, and destroy it by forming new compounds with it, thus forming one kind of corrosive action; while, for bilge-water, another kind of corrosive action is added, thus accounting for the variety in the corrosive phenomena, and for the difference in the rapidity of the action. There may, also, be stated that nearly all vegetable matter contains a little sulphur in some form dissolved in its sap and incorporated as foreign matter in its fiber.

The hydrogen and oxygen of cellulose, when the cellulose decays and disintegrates, may decompose from their combinations and the former may unite with the sulphur of the sulphur salts dissolved in sea-water, and, in some cases, with the chlorine of the common salt dissolved in sea-water. Further, the oxygen constituent of the cellulose, liberated in the nascent state furnishes an extremely efficient corrodent. Some electrical action no doubt is likewise a factor.

The whole subject is very obscure and complex, but not so confused that a tolerably accurate idea cannot be formed of the causes and consequences of the corrosion in question.

(*a.*) The corrosion of the metal is in all cases more or less rapid as it is more or less impure, not only because the impurities may be more easily acted upon, but also because by separating the molecules of the pure metal farther apart, its cohesive strength is thereby correspondingly impaired.

(*b.*) As regards the temperature at which the corrosion takes place, the higher the temperature of the metal, the more rapid is the rate of the corrosion. This result is doubtless due to the fact that the density of the metal decreases as its

temperature increases, its molecules being kept at a greater distance apart with higher than with lower temperatures whereby the force of the intermolecular or cohesion attraction of the metal is lessened in some ratio to its increase of temperature, and its resistance to chemical combination with other substances is thus correspondingly lessened.

(*c.*) The larger the mass of the metal exposed to corrosion, other things equal, the smaller will be the rate of its corrosion; that is to say, a less absolute weight of metal will be destroyed the greater its mass.

In the case of condenser tubes, or of the metal sheathing of wooden ships, the thicker the metal the slower will be the rate of its corrosion, and not only relatively but absolutely. As regards the condenser-tubes, the thickening of their metal involves not only increased money cost at first, but increased weight to be permanently carried afterwards, and somewhat increased dimensions of the condenser. This money loss at first, however, is recouped at last by the greater durability of the tubes above the proportionality of their increased weight, leaving out of consideration the valuable convenience of their longer endurance which is a merit of importance especially for naval steamers. The thickness now used for the metal of the tubes should be doubled.

The gain of resistance to corrosion in the case of increased thickness of metal, seems to come from the fact that the action of the corroding power is limited to the surface exposed, while the resistance to it is, in some ratio, proportional to the mass. The combined resistance of the intermolecular attraction of the mass seems to be a factor in lessening the corrosion in a higher degree than the increase of the mass.

(*d.*) Coating the tubes with tin delays their corrosion as long as the tin lasts, but its durability is not great, and its very slight thickness on the tubes cannot be increased beyond what is obtained from the first "dip." This very small thickness of tin prevents the action of the tube-metal from taking an increased thickness of tin from additional "dips;" and it also diminishes to an appreciable degree the heat conducti-

bility of the tube; nevertheless, this "tinning" should never be omitted, although it involves additional surface of tubes for equal heat-conduction in equal time.

(*e.*) In surface-condensers the temperature of the exhaust-steam is always higher than the temperature of the metal of the tubes in contact with it; and the temperature of the metal of the tubes is always higher than the temperature of the water of liquefaction in contact with it. This water, which is the normal feed-water, has, therefore, in the condenser in a marked degree, a temperature inferior to the temperature of the steam from which it is precipitated. Statically, such results would be impossible, but the case is one of dynamics, the heatings, coolings, and replacements being successive and time not being sufficient for obtaining equilibrium.

(*f.*) The air-pump draws from the surface-condenser not only the feed-water but also the unliquefied, exhaust-steam; it draws them simultaneously, and delivers them mixed together into the hot-well, consequently there occurs in this operation an impartation of heat by the exhaust-steam to the feed-water, so that the temperature of the latter in the hot-well may be higher than in the condenser. Furthermore, the air-pump compresses the exhaust-steam from its pressure in the condenser to at least the atmospheric pressure in the hot-well, if the hot-well be open to the atmosphere, thereby increasing the heat in that steam by adding to it the heat equivalent of the work of the compression.

(*g.*) The corrosion of the condenser-tubes is principally on their sea-water side, but their distilled-water side also suffers much, the corroding agents being the atmospheric gases mingled with the steam. This corrosion is quite uniform compared with the corrosion on the sea-water side, and the resulting oxides and salts are often carried into the boiler with the feed-water, and produce injurious effects upon the iron.

(*h.*) The more rapid the current of the sea-water over the surface of the metal, the more rapid will be the corrosion, as the oxides and salts of the corrosion having but slight adher-

ence to their metallic base, will be more speedily and thoroughly washed off, leaving the raw metal to be acted on.

(*i.*) In the case of horizontal condenser tubes, there will often be found a partial coating on their upper sides of baked mud, sometimes as hard as brick. This is mud deposited by gravity from the sea-water on the upper outer side of the tubes when the sea-water surrounds the tubes, and on the inner lower side when the sea-water passes through the tubes, and has been baked by the long application of the low temperature of the tubes. It is often mistaken for sea-water scale, but its nature may with certainty be known by mere inspection from the fact that it is not found on the lower outer side, or on the upper inner side of horizontal tubes, according as the sea-water is without or within them, nor on the outer or inner sides of vertical tubes. It is a great intercepter of heat and reduces correspondingly the steam-liquefying power of the tubes.

(*j.*) A surface-condenser should be opened at intervals, say, of six months; its tubes drawn out, cleaned, and examined with the particular view of ascertaining if any holes have been corroded through them by means of which the sea-water could find entrance into the distilled-water space. If any sea-water, however small in quantity, thus mingles with the distilled-water feed, its mineral matters held in solution will be precipitated upon the water-heating surfaces of the boiler in the form of scale. Such leakage will be shown by the "salinometer" if one be used, and one should always be in use, as it will indicate the gradual salting up of the water in the boiler even if the latter leak so that no undue accumulation of water in the boiler be apparent. Now, as there is always deficit of distilled water in all boilers under all circumstances, which deficit is restored by the "evaporator," a certain amount of leakage of sea-water into the distilled-water space of the condenser may not be shown by a rising water-level in the boiler, but will always be shown by the increasing density of the water in the boiler made evident by the "salinometer;" it will also be shown by the lessened quantity of distilled

water required to be furnished by the "evaporator." The presence of even a small proportion of sea-water in the water in the boiler can be discovered by a delicate taste, and also by any of the usual chemical tests. But the fact that the contamination of the distilled water in the boiler by inleaked sea-water is due to the perforation of the tubes in the condenser by corrosion, can only be shown by ocular demonstration.

THE PRESENT STATUS OF MARINE GAS ENGINEERING.

By Peter Evermann, Associate.

[Continued from Vol. XVII, No. 2, "The Future of Marine Gas Engines."]

In presenting this second paper on Marine Gas Engines to the Society, I shall deal with the internal-combustion engine as it appears today in all kinds of services on salt water as well as on fresh-water vessels.

In my first paper I treated more particularly of the *future* development of the gas engine, and referred to the theory of the gas producers. In this paper I shall have to consider the *present* status and some details in regard to gas plants running on liquid fuels as well as on solid fuels, and also propellers and engine connections. Some recent lectures, as they appeared in Europe, and parts of which are translated, gave some interesting additions to this paper.

The development which the gas engine has undergone in the past ten years, especially in regard to reliability and the fuel-saving question, has influenced and encouraged both the builders and owners of vessels, and I think it worth while to present it in this paper. The first engines running in boats have been using illuminating gas. This gas was stored in big tanks under high pressure, as it was impossible to install an illuminating-gas factory on board. But the tanks required such a large amount of room and weight that they seriously influenced the tonnage of the vessel. This is the main reason why only a very limited number of boats driven by city-gas engines have been constructed.

In Paris there was a freight boat, *Idée*, running in the year 1895 between Havre and Rouen. The engine was of the two-cylinder vertical type, and is shown in Fig. 1. It had 10 H.P., and the fuel, city gas, was stored in 80 tanks, with

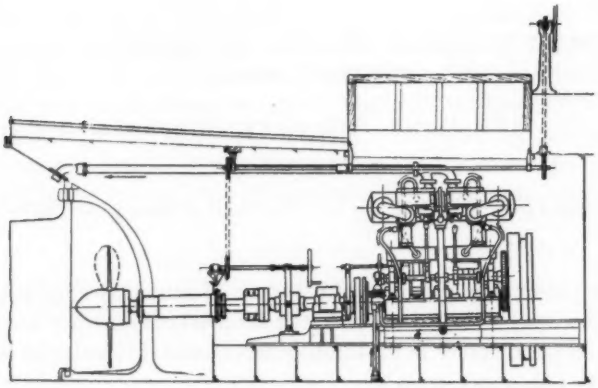


Fig. 1.

1,500 pounds pressure per square inch. Each tank had about 800 cubic feet capacity; therefore there were on board about 64,000 cubic feet. A good modern gas engine consumes about 16 cubic feet per H.P. per hour; thus, the steaming time of this ship was 100 hours at 6 miles per hour, and gave a radius of action of 600 miles. Each tank weighed 715 pounds, and the vessel was therefore carrying a dead weight of 57,200 pounds.

A good kerosene engine will require about one pound of fuel per H.P. per hour, or 40 pounds for a 40-H.P. engine. It will therefore be seen that this boat, when using kerosene, will have an action radius of 7,200 miles, if we assume the weight of the fuel tanks at only $4\frac{1}{2}$ tons, which is a fair estimate.

It was not surprising that the illuminating-gas engine was soon displaced by the developing of the oil engines. The latter were independent of city-gas plants and compressing stations, and much cheaper in service.

The gasoline engine in the United States, and the kerosene engine in Europe, have been greatly developed since, and other liquid fuels, as alcohol, crude oil, etc., are also helping to supplant the steam engine.

One hundred and fifty years of marine steam engineering give, of course, some very valuable information as to "how to build a good marine gas engine." No reliable concern of today has had good luck in placing a stationary steam engine in a boat, or in using a marine steam engine for a stationary electric-power plant. It is useless to try this in gas engines, where the question becomes more complicated, from the simple rule, that each gas plant should be built to suit the specific fuel, and not a specific engine type suitable for all fuels. More and more are direct reversible marine gas engines being placed on the market, and the devices in use now, such as reversing coupling or reversing propeller, will disappear in the larger units.

A very light engine with a very high number of revolutions is the explosion engine for light boats and launches, and this is the marine engine of the present. Reliable, heavy engines, with thousands of horsepower, have not yet been built. Boat motors of today have 20 to 100 pounds per H.P., and stationary reliable engines have from 200 to 500 pounds per H.P. and more.

As the usual four-cycle, single-acting engine with only one cylinder is not at all equal, in regard to steadiness and vibrations, to the steam engine, two, three and up to six and eight cylinders have been installed, thus obtaining a motor which works very steadily, and is satisfactory as regards the balancing of the moving parts.

Experiments in this regard, as using two pistons in one cylinder, have been made, but proved to be not successful and too complicated. The cylinders of these have all been horizontal. As the greater number of gas engines today are non-reversing, I will first briefly touch on the required details of reversing propeller and reversing coupling.

It may be noted that in Europe the reversing propeller is just now more in favor than the reversing shaft coupling. Such a coupling is shown in Fig. 2; (a) is the rear crank of the gas-engine shaft, (b) the flywheel, (c) the propeller wheel, (d) the thrust bearing, and (e) the reversing clutch. A stronger

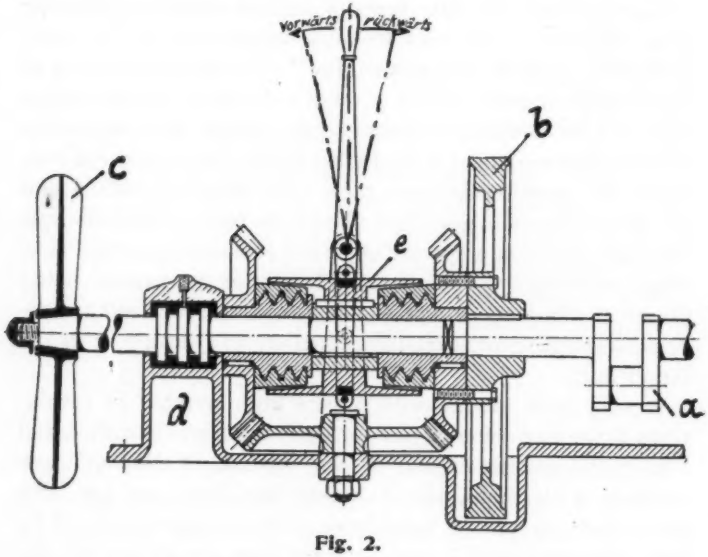
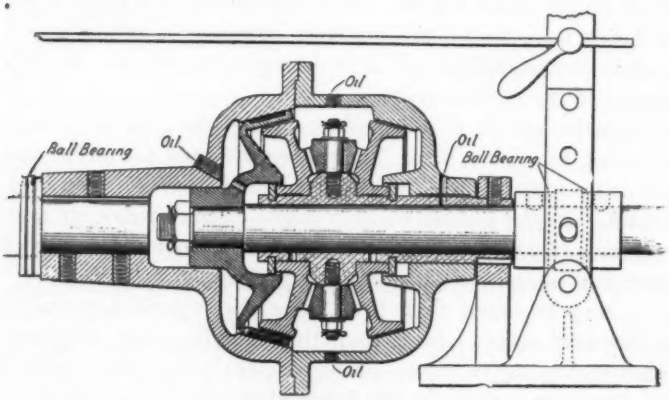


Fig. 2.

reversing coupling is shown in Fig. 3. The thrust of the propeller is directly on the friction cone, and sudden shocks on reversing are nearly avoided. All parts are enclosed in



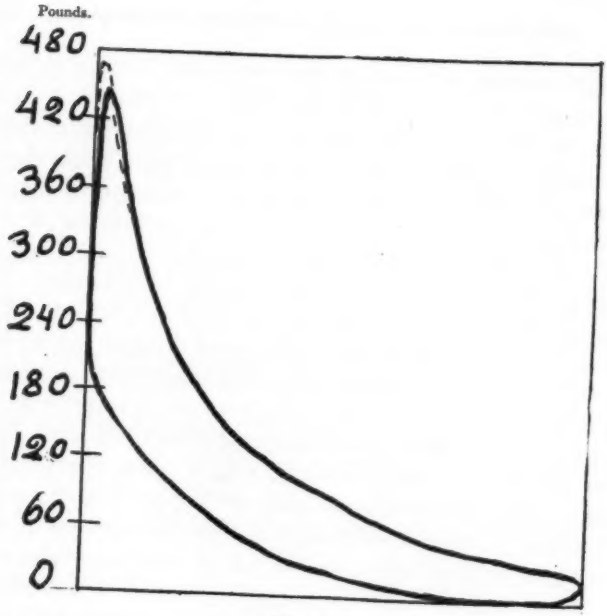
REVERSING GEAR.

Fig. 3.

oiltight casings, and it takes up a very small amount of room. The weight of it, for a 2¾ shaft, is only 725 pounds.

In the last few years another type of reversing coupling has been developed—the so-called coil-spring friction clutch. These clutches are in use for 2,000-H.P. rolling-mill engines, and they will work well in boats. A picture for a boat outfit was shown in "Iron Age," December 14, 1905, and the arrangement for a 800-H.P. steam engine is as follows: The engine has about a 10-inch shaft and 80 revolutions. This speed is cut down to 40 revolutions, and the gear works on a 15-inch shaft. It is interesting from the marine engineer's view to know that the engine has a 46-ton flywheel near this clutch. The coupled continuation of the engine shaft has a spur gear with 27 teeth, meshing with a 54-teeth gear on the clutch shaft, and a 28-teeth gear on the back shaft; on the other end of the back shaft meshes a 45-teeth gear with a 90-teeth gear on the clutch shaft. All of these are spur gears and have a pitch of about five inches. It will be noticed that the two spur gears on the clutch shaft are arranged to run at about 40 revolutions, in opposite directions. By means of one or the other of the two coil clutches either of these may be engaged at will with the shaft. The thicker end of each coil has a lug formed on it resting in a pocket at the side of one of the gears, while the other end is connected to the short arm of a bell crank on the same gear. When the flanged collar is moved it comes into contact with the bell-crank lever of the corresponding coil, and the last turn of the coil is tightened on its drum, so that the entire clutch grips with great power

A good reversing coupling should have, however, the advantages of all spur gears, and that it occupy a relatively small space. In backing the speed is less, and the work for the gas engine easier. It is usually not desirable to run the propeller for backing as fast as for ahead, on account of the higher friction. This fact is sometimes the cause for sudden shutdowns of the engine, and is especially bad in smaller craft, where an easy handling is often necessary. It is worth while to note here that the usual gas engine of today reduces


Pounds.
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420
360
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240
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Fig. 4.

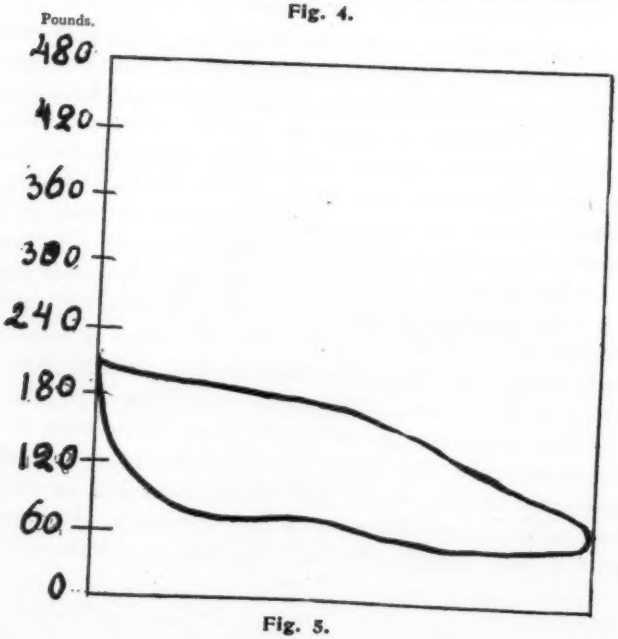
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Fig. 5.

its energy and power with the reduction of revolutions, thus acting just contrary to the steam engine, where, with larger openings of the valve gear, even more power is obtainable under fewer revolutions. The slowing down of the vessel requires less revolutions, and under these conditions the gas engine works very uneconomically, as the exploded mixture has too much time for the internal losses, due to the cooling influence of the water.

Fig. 4 shows the diagram of a high-compression gas engine, and Fig. 5 the one of a high-pressure cylinder of a marine steam engine. The steam diagram may have about 90 pounds mean pressure and this gas engine diagram rather less, in spite of a six-times higher initial pressure for the gas engine. The three-throw crank shaft has about 14 inches diameter for a 2,000 H.P. steam engine; this would be, however, just sufficient for a gas engine having about half the power, namely, 1,000 H.P. or 1,200 H.P.

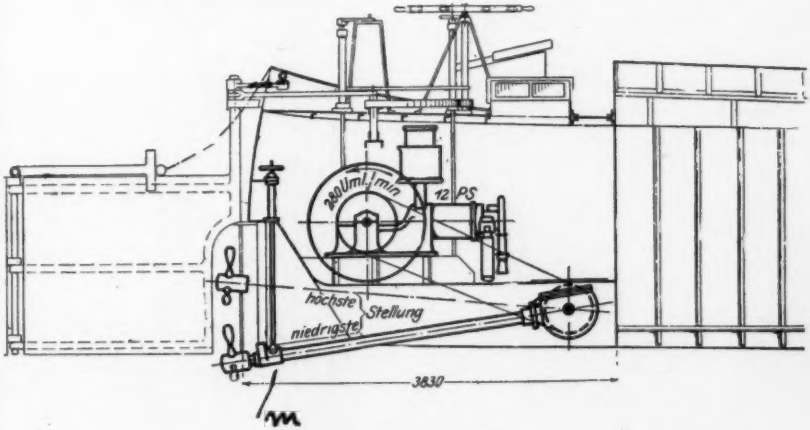

Fig. 6.

The second way of reversing was found in constructing reversible propeller wheels in connection with the non-reversible gas engine. A very interesting solution of a reversible propeller is shown in Fig. 6 and Fig. 7. The condition was to install the gas engine in a boat having 12 inches draught

at no load and 72 inches at full load, with a speed of about 1½ miles per hour. The latter was allowing a small propeller wheel, but the great difference in depth, at first, caused considerable difficulty. A solution was found in combining a liftable propeller wheel with a reversing mechanism. The force is transmitted by the shaft (a), through the rotating gear

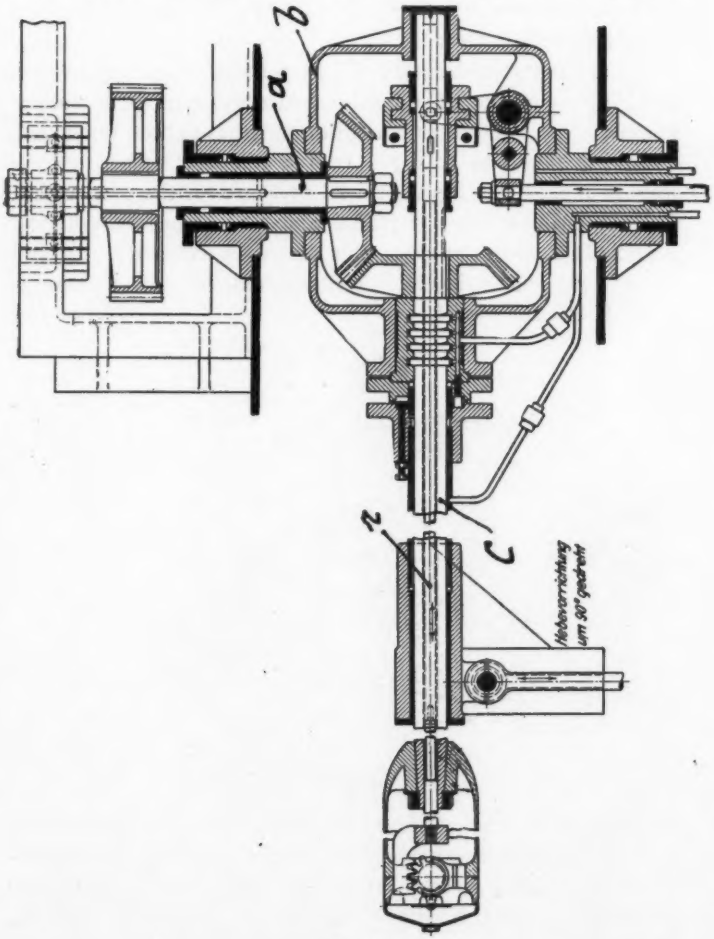

Fig. 7.

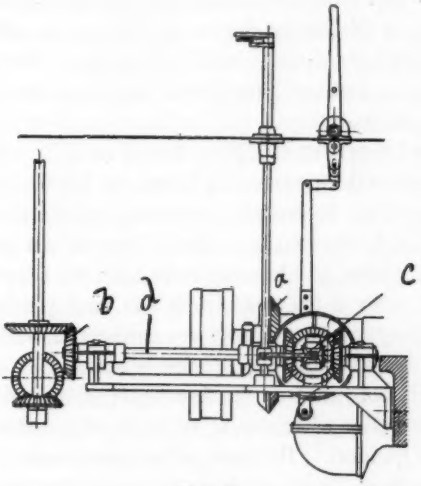

Fig. 8.

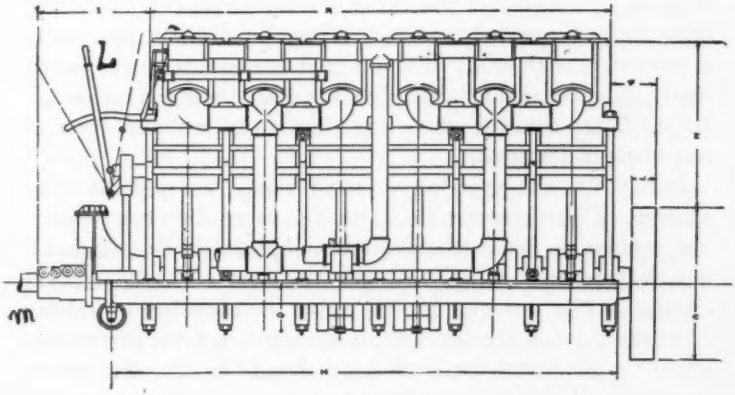

Fig. 9.

casing (b), by means of bevel wheels to the propeller shaft (c), as shown in Fig. 7. The thrust bearing acts on the two casing bearings, thus transmitting the engine power to the ship. The reversing of the wheel is done by a rod (r), inside of the propeller shaft, and the blades are turned by teeth, acting like

spur gears. - The entire mechanism is enclosed and safe. The arrangement of lifting is shown in Fig. 10, at the stern, by means of a handwheel, screw and rod, at (m). The gas engine in this case is a standard, horizontal, stationary one, and belted to the main drive.

As soon as the power is higher than 100 H.P. it is often very difficult to move these reversing levers by hand alone. It was therefore required to install operating mechanisms for this purpose. Fig. 8 shows the vertical view of an arrangement whereby the power of the engine is used to reverse the propeller wheel. By a friction clutch (a), and gearings (b), and a claw coupling (c), those parts are connected, and act on the inside-shaft movable rod (d) for the blades.

Here I will note that in the first years of their development reversible propeller wheels were strongly objected to by marine engineers in general. But now, after about twenty years' experience on these lines, it may be said that most factories have very little trouble due to defect in weak construction or to wear. The greater portion of the complaints is due to ice or to running aground, and here it may be said that just such a propeller, with easily disconnected blades, is much more easily repaired, and at much less expense, than solid wheels. In the latter case the entire screw has to be renewed if only one blade is damaged.

In regard to the reversing question it may be noted that the number of direct reversible engines is increasing every year, and there is no doubt that great development in this important feature will follow, as was the case with the steam engine. The gas engine will never be considered perfect from the view of the Navy engineer, if it is not directly reversible. A good deal has been done already in this direction, however, and, besides some European concerns, there are some very good American makers coming in the market. The later makes are, in some cases, of the 6 and 8-cylinder type, and they have some advantages. Vibration is nearly eliminated, and, owing to the increased number of cranks, these machines run as smoothly as the finest balanced steam engines. The

flywheel and weight are greatly reduced, as the torque and strains are almost continuous. If desired, the engine can be operated without a flywheel. An elevation is shown in Fig. 9. Starting and reversing is as positive as in a steam engine, and the propeller is coupled at (m) direct to the engine shaft.

The operation is as follows: The valve motion is shifted by the lever (L) from a central point to the forward or reverse motion, and on opening the compressed-air valve, the engine, having no dead centers, instantly starts up. The compressed air is then shut off. A small air compressor on the engine keeps up the supply of air. The steel tanks hold enough air for starting 20 or 30 times without pumping. In the very large engines there are auxiliary air-pump engines, which are similar to the auxiliaries supplied with a steam outfit. A number of Russian torpedo boats, with 1,500 H.P., have been equipped with these famous engines.

Another method of reversing the motion of the engine is by advancing the spark, thereby causing a so-called "kick back." This feature is reliable enough for smaller boats. These engines with low compression can also be started ahead or astern at the will of the operator.

Like all other igniter operating devices, this one is equipped with devices for changing the time of ignition, according to the quality of the fuel, as well as according to the speed at which it is desired to run the engine.

In starting the engine the reverse lever is placed at a point which sets the igniters in such a way as to make the spark, after the piston starts on the downward stroke, in the direction it is desired. In reversing, the electric-battery main switch is opened, and the lever moved across the quadrant to the opposite side. The switch is closed after reversing.

As was natural, the gas engine was first introduced only in small sizes and in small craft. One of the earliest gas engines ever installed on water is probably the city-gas engine operating the floating bridge over the Rhine, in Cologne, since the year 1882. This was a four-cycle Otto engine of special design, with vertical shaft; this type of engine has since been

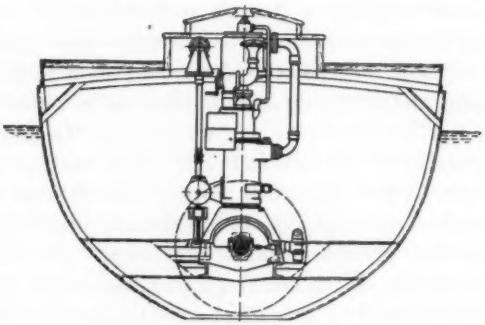

Fig. 10.

abandoned. The four-cycle principle, however, is leading to-day in marine gas-engine building. The two-cycle principle is not yet so well developed.

An early two-cylinder vertical kerosene engine of foreign make is shown in Fig. 10. It shows the lighter *Esse* with a 16-H.P. kerosene engine. The boat has a speed of 8 knots. The engine shaft is coupled to the propeller shaft, and the screw blades are reversible. The operating wheel is on deck to the right of the steering wheel. It is interesting to note the relative costs per ton, mile and hour. At a kerosene consumption of about .88 pounds per H.P. an hour, and a price

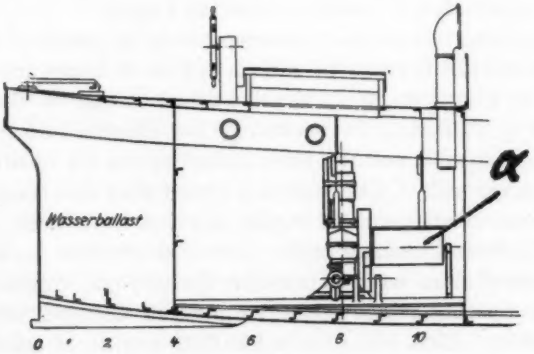

Fig. 11.

of 2.3 cents per pound, the cost per one ton, knot, hour, for 40 tons load was: $(16 \times 0.88 \times 2.3) : (8 \times 40) = .10$ cents. In reducing the speed to only $4\frac{1}{2}$ knots and using only a 6-H.P. engine the cost would be: $(6 \times 0.88 \times 2.3) : (4.5 \times 40) = .067$ cents. At that time marine gas-engine practice was not so good as today, when it is known that there must be a proper relation of speed and load to obtain the best and most economical result.

In 1896 the *Petrolia* was equipped with kerosene engines. She was built as a sail boat, having kerosene tanks of 210 tons capacity. She had a length of about 100 feet, beam of 18 feet, and 7 feet draught. The engine developed 20 H.P. at 360 revolutions per minute; the arrangement was in principle the same as in *Esse*. The cost per knot, ton, hour, has been reduced to $(20 \times 0.88 \times 2.33) : (210 \times 4) = 0.048$ cents, with a speed of about 4 knots per hour. It must be noted, on the other hand, that the expenses for the wages of the crew have to be considered as well. A similar early towboat has 50 feet length, 10 feet beam, and 5 feet draught, and is equipped with a 20-H.P. kerosene engine. The speed is about $8\frac{1}{2}$ knots per hour. The propeller wheel has two blades, and is 3 feet in diameter. There is a tank of about 40 cubic feet capacity for the supply of the engines, which is sufficient for about 100 hours' run, or about 850 knots action radius for only 40 cubic feet of bunker capacity. The entire machinery equipment requires about $\frac{1}{7}$ of the ship's length. An equivalent powerful steamer equipment would need about $\frac{1}{3}$ of the ship's length. With a coal consumption of $4\frac{1}{2}$ pounds per H.P. per hour, which is certainly fair, the comparison shows as follows: the above 40 cubic feet would contain about 2,400 pounds of coal, which is sufficient for only 26 hours' run, or about 221 knots action radius. Considering, however, the costs per H.P. hour, the comparison would stand as 1 to 2 or 1 to 3 in favor of the steam power. But as the action radius stands as 4 to 1 or 5 to 1, so this example shows characteristically the great value of small motor boats as auxiliary craft to be kept on board of large men-of-war, and to be used in cases where quick action is required.

The arrangement of a foreign 4-cycle gasoline engine driving twin propellers is shown in Fig. 11. The two propeller shafts are each driven by a pair of open and crossed belts from a main pulley connected to the engine shaft. This arrangement, however, has proved to be not very satisfactory, as the belting on such short distances of the shafts did not work well.

I will state here that it is advisable, for various qualities of liquid fuels, to use various arrangements of devices and methods to form the mixture in explosion chambers. Obvious, however, as this rule might be, there are numerous mistakes made. It has not been, and never will be, practicable to use the same devices for various fuels. One physical rule is to be observed for all similar combustions: separate the fuel in as fine and small particles as possible, and mix it as thoroughly as possible with the air. One of the best authorities, C. Stein, Germany, in this respect, says: "It is nonsense to attempt to build all marine combustion engines of a standard type. Each experienced engineer must know that the conditions in service for pleasure launches, tow-boats or naval purposes are entirely different. It would be certainly agreeable for the manufacturer to use one pattern for all systems, but it is wrong, as it will never suit the customer."

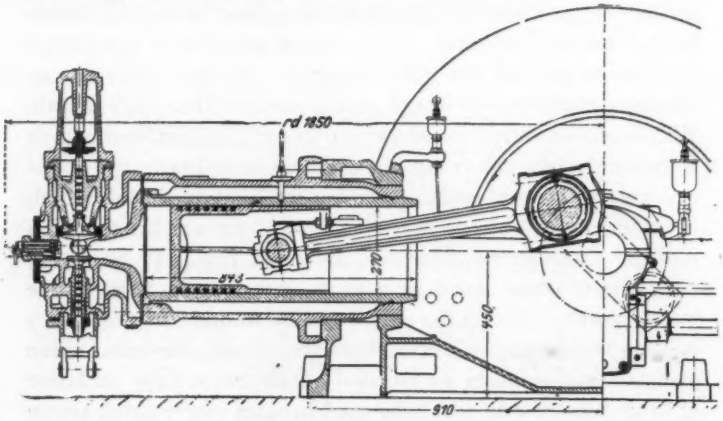

Fig. 12.

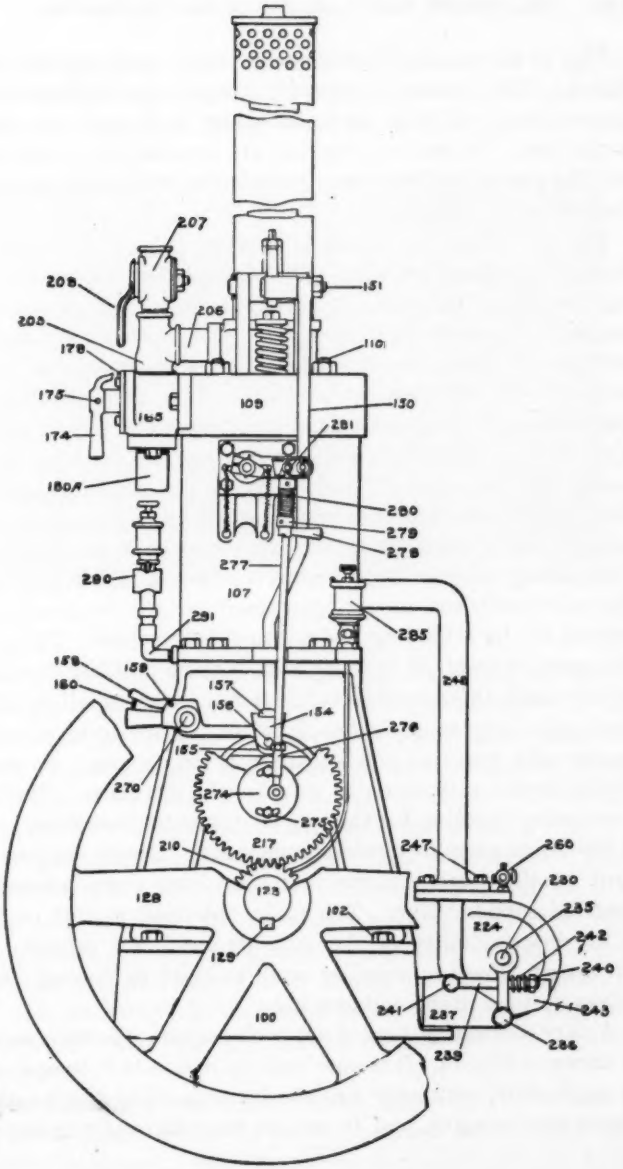
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Fig. 13.

Fig. 12 shows a horizontal marine engine with opposite cylinders. This pattern is specially suitable for high-compression engines, and is to be made lighter or heavier, according to the fuel. Valves and igniters are mechanically operated, and the piston can be drawn, towards the crank shaft, without disturbing any other part.

Fig. 13 shows an American engine of very well known make. It is based on the 4-cycle principle and has the following important features: A check valve in the air-suction, automatically-working suction valve, and mechanically-operated exhaust valve; an independent throttle arrangement for each cylinder, operated by hand as well as by a governor, a spark-timing device, and a hand-speed controller for the governor. It is a specially good gasoline engine, and the air is heated by the exhaust before starting the mixture process. The exhaust can be made upright, as shown in Fig. 13, or submerged, but is provided in the latter case with an air check. The cooling water is furnished by a rotary pump driven from the main shaft, and an air compressor for the required compressed air for whistling and starting, is attached. The electric spark is obtained by battery or dynamo. When distillate is to be used, there must be two fuel tanks—one small gasoline tank and a larger one for the distillate. The engine must be started with gasoline and should run long enough to warm the air heater sufficiently to vaporize the distillate. There is a reversing coupling for the propeller, and the operating lever is similar to a steam-engine arrangement. Larger engines are built on the same principle, but they have a mechanically-operated suction valve. The main objection to this engine is the inaccessibility of the cylinder heads for repairs; an advantage, however, may be said to exist in having both valves vertical in the cylinder head.

A very interesting type of a 4-cycle engine, of foreign make, is shown in Fig. 14. It is now built up to 600 H.P. in one unit of 4-cylinders, with only 120 revolutions. The single-acting piston was accepted, and it reduces the weight of the engine.

High compression was preferred, as it is built as a special marine producer-gas engine. I will say here that I perfer this system for very clean liquid fuels, but not for producer gas.

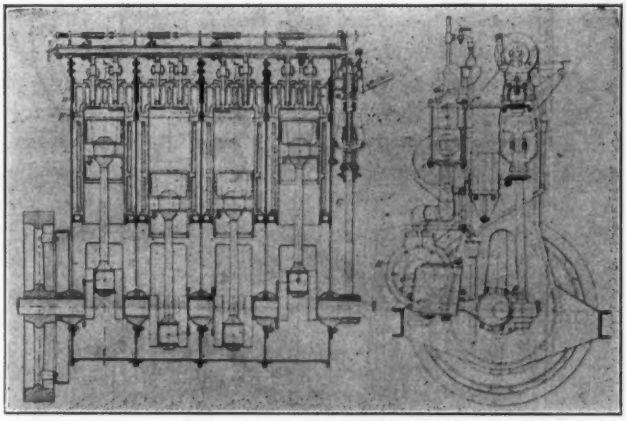

Fig. 14.

The engine shown has no cages for the valves and it is, therefore, necessary to remove the entire cylinder head if the valves must be cleaned. The cylinder hangs down from the heads, thus allowing a free expansion, and the strain of the explosions is taken up by a steel-plate structure. These are very remarkable features, but only time will prove if they are really good for marine purposes. The countershaft runs across the heads and operates the valves by single links, and the igniters are timed by means of a rod lying in this hollow countershaft. This arrangement, I think, is especially preferable in smaller engines, as it allows an easy accessibility, and observation of the operating parts. For larger engines, however, it will probably be objectionable on account of wear and noise.

For operating a boat in shoal water, however, it is necessary to provide other means for propulsion, and stern wheels are therefore preferred. Side wheels are used principally for ferry

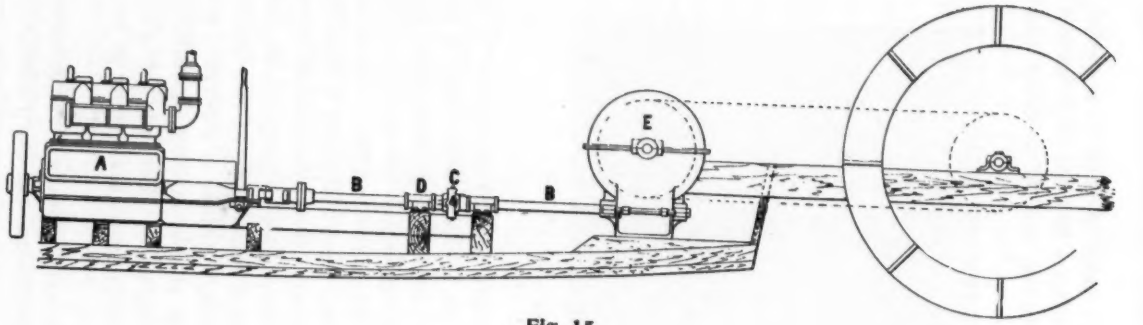

Fig. 15.

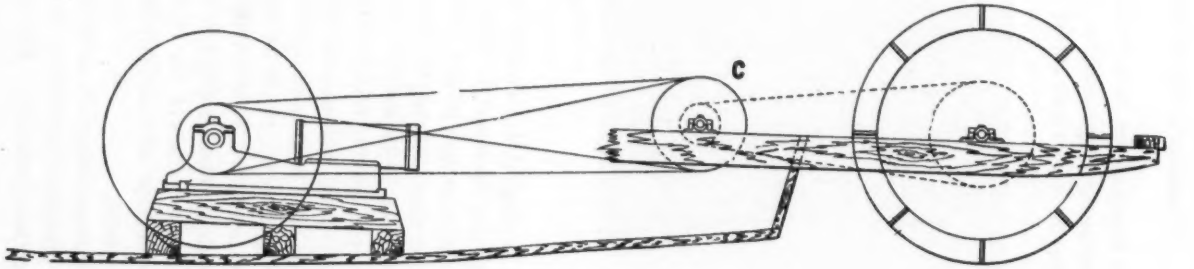

Fig. 16.

boats where it is necessary to provide a clear space through the center of the vessel, and to make a landing with either end at the dock or slip. Fig. 15 shows the arrangement of a vertical gas engine, where the power is transmitted to the stern wheel by means of a worm-gear and sprocket-chain connection. Fig. 16 shows the installation of a standard horizontal stationary engine in a boat, transmitting the power to the stern wheel by means of straight and crossed belts, and tight and loose pulleys, to a countershaft F; thence to the wheel H by a sprocket-chain drive L. The reversing is done in case 41 by a reversing clutch, and in 42 by the crossed belts.

A characteristic American construction is shown in Fig. 17.

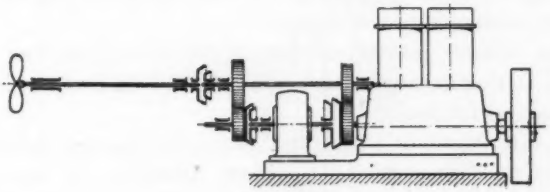

Fig. 17.

It is a 200-H.P. kerosene engine, vertical, with two cylinders, and especially designed for submarine-boat services of the Holland type. Directly connected to the engine shaft is an electric generator, thus taking advantage of the high speed. The screw propeller shaft is driven by spur gears. On the surface the engine runs the propeller or charges the storage batteries. Below the surface this generator is switched in as a motor and drives the propeller wheel from the battery. This arrangement is also very convenient for starting the kerosene engine by the electro-motor.

Another citation of authority, Rankin Kennedy (England), on marine engines is as follows:

"The moving parts of the engine should be the same as those of a standard marine *steam* engine, and there is no reason why they should be otherwise. To box them up, *out of sight*, in a crank case, would be *absurd*, although among oil and kerosene motor-car engine makers this proceeding is *fashion-*

able. When we come to marine (navy) engines, the conditions are so far different that *open* engines must be insisted upon, and other departures from the conventional practices must be made when marine propulsion by internal combustion engines is *seriously* considered, either for commercial or naval purposes. The present craze is to install too light weight motor-car engines. This is not good marine engineering. As a matter of safety the marine engine is far too important to be trifled with, and experience with the steam engine is not to be lightly thrown aside for the whims of the fashionable motor-car builder. Hence in designing internal-combustion engines for business purposes the closed motor-car model is useless, and engines modelled more in the shape of steam engine practice are necessarily to be adopted."

That *reliable* gas-engine construction was going the same way as stationary engines may be proved by a few words and drawings as follows:

Fig. 18 shows a "Korting" two-cycle engine, driving a horizontal blower engine for blast furnaces. Is there any service in mechanical engineering where *reliability* and *durability* is of more importance than in the blast furnace practice? These engines must run years and years, without giving any considerable trouble, and yet we see the steam engine replaced by the gas engine.

Fig. 20 shows the "Deutz" four-cycle engine, and if the piston rods and valves did not show "water cooling" it might be easily mistaken for a modern steam engine.

Fig. 19 shows the "Cockerill" tandem gas engine, especially as twin-tandem, giving any degree of steadiness, and satisfying the most severe conditions of running alternating generators in parallel.

Fig. 21 shows another big engine, "Oechelhauser," two pistons, single cylinder, two-cycle, well known on account of its claim as a valveless engine, specially preferred for heavy services, like rolling mill and blower engine driving.

Fig. 22 shows the "Nurnberg type" driving a blower engine, but with specially strong claims as being an engine of good regulation, and steadiness, and famous workmanship.

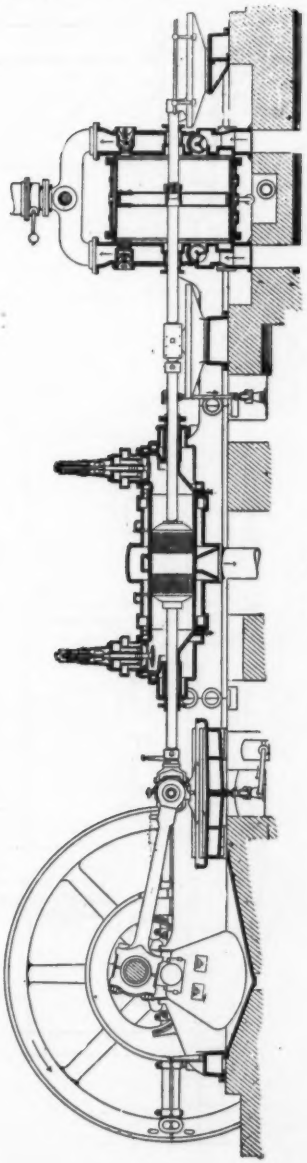

Fig. 18.

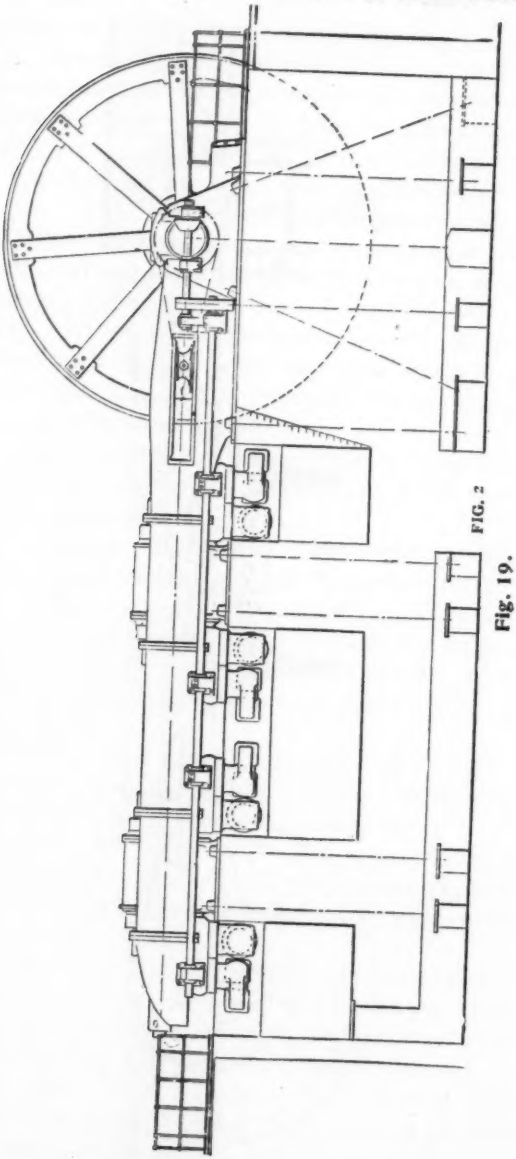

FIG. 2

Fig. 19.

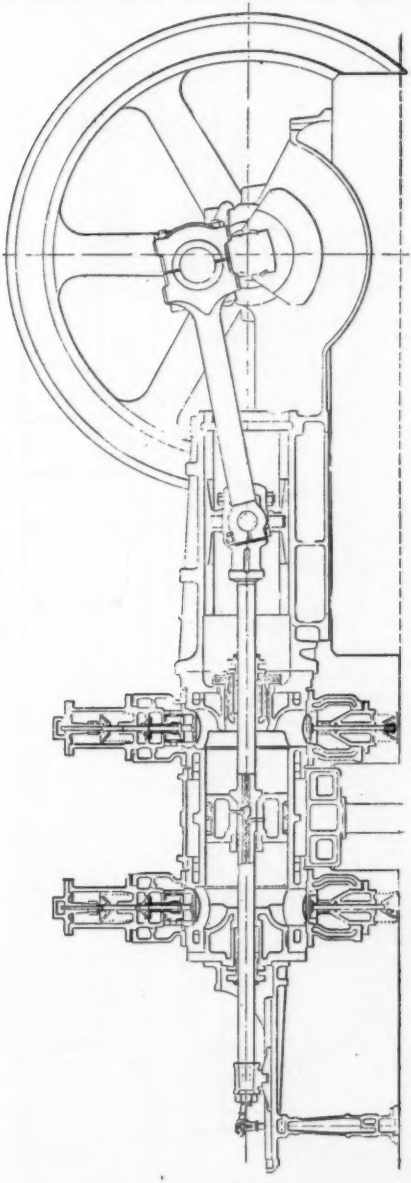

Fig. 20.

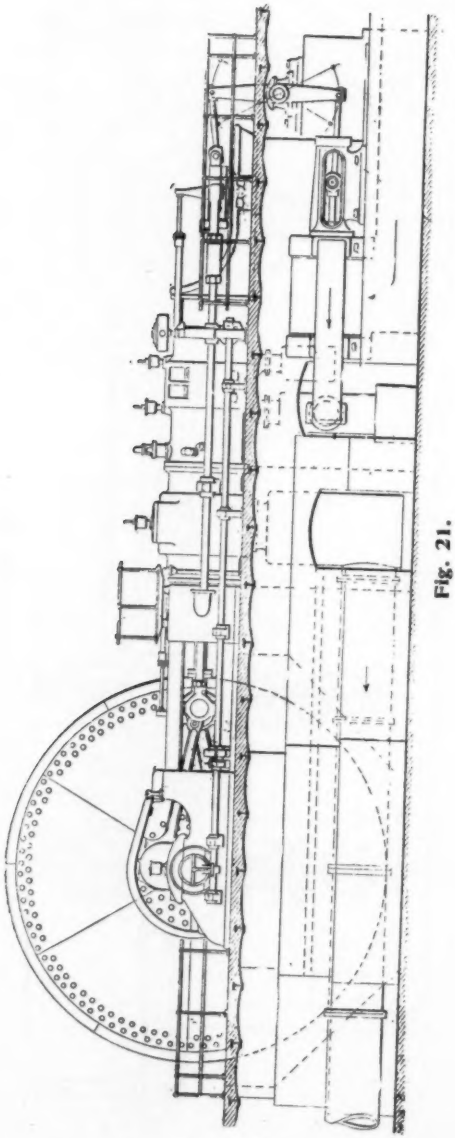

Fig. 21.

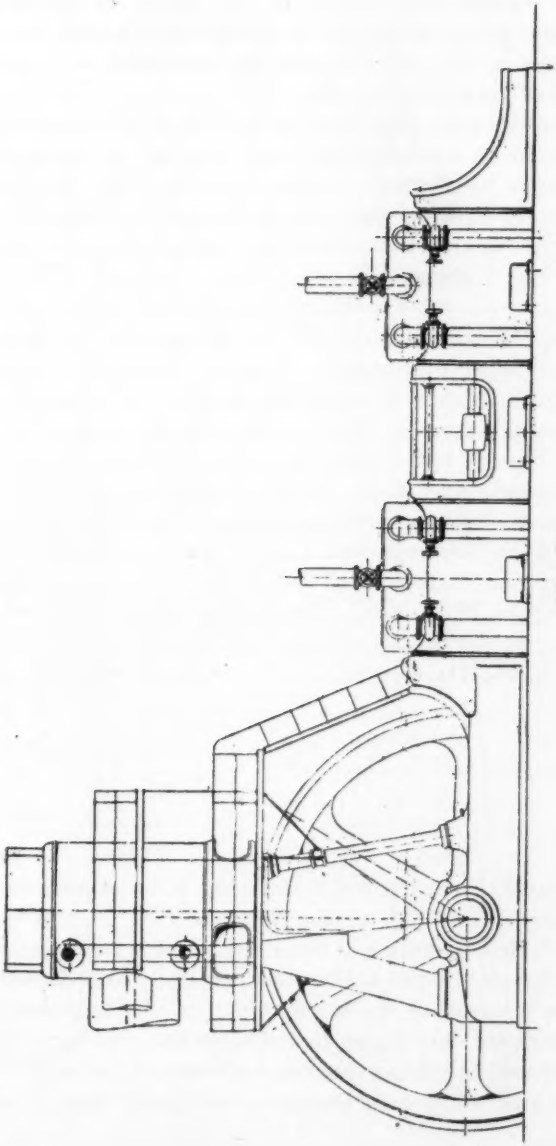

Fig. 22.

This greatest development in the history of mechanical power was giving confidence to the marine engineer, and the latest step in this direction was the installation of "suction-gas plants" in marine vessels.

Before entering, however, this field, there may be said a few words more on modern power boats running on liquid fuels. It might be noted that in the last few years there has been a tendency to construct the hull of the vessel to suit the conditions of the internal-combustion engines, and this type of boat is very different from those driven by steam. Therefore it would be advisable to follow the practice of steam engineering so far as to furnish not only the gas engine, but also the boat and other equipments. Usually, a shipyard furnishes both the hull and the entire machinery. Why not do the same in gas engines? This is done now, more and more, and a new type of hull construction has resulted. Tests with small models were made by Mr. Kretschmer, of Berlin, and gave the following interesting results: The model was made according to the dimensions of an up-to-date ocean liner, about 630 feet long, 75 feet beam and 24 feet draught, and having a tonnage of 16,800 tons. The corresponding results of this model with the new tetrahedral form are given in this table:

Speed and knots per hour:	Required horsepowers:
16.3	4,150
19.6	7,450
22.8	13,120
25.0	19,000
26.1	22,500
29.7	37,900

A boat of French make of this kind is to be seen in Fig. 23, *Titan II*, and Fig. 24, the *Rapee III.*

The widest framing is at the stern. The bottom portion of the hull is ball shaped and the bow is entirely flat. In motion, the bow rises out of the water, which results in a decreased water friction. Bow waves are not noticeable, and there is only a slight swelling of the water on both sides of the stern. The length over all of both the above mentioned boats is about

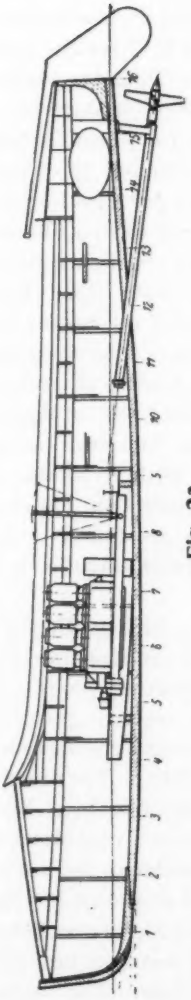

Fig. 23.

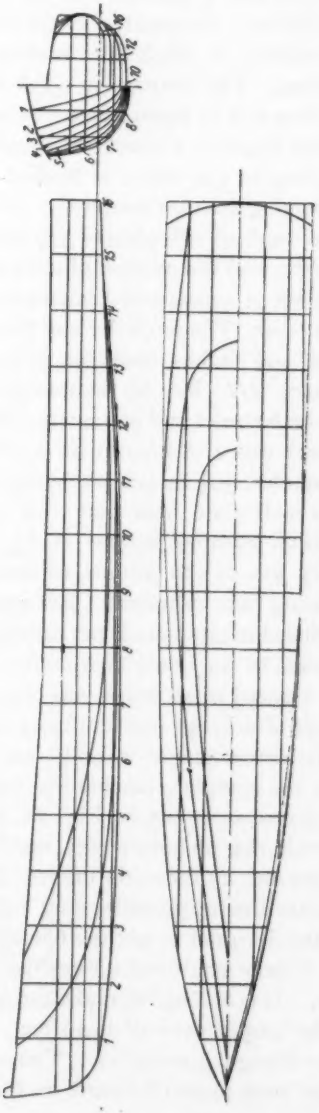

Fig. 24.

26 feet and 3 inches, and the length on water line is 25 feet 10 inches; the greatest beam is 5 feet; the draught, at rib, is 6¼ inches, at rib 8 is 11 inches, and at rib 16, at stern, is 22.5 inches. The tonnage is 1,158 and the largest cross-scantling section is 2.15 square feet. It was necessary to drive the propeller shaft by a universal coupling, in order to get the blades as deep in the water as desired. The reversing is done by a reversing friction coupling. The oil engine for *Titan II*, has four vertical cylinders of $5\frac{5}{16}$ inches in diameter and 5⅛ inches stroke, and the number of revolutions is 1,415 per minute, giving about 54 indicated horsepower. The speed was 19.2 knots per hour. The propeller had three blades with $18\frac{7}{8}$ inches pitch and 20¾ inches diameter, giving 12.3 per cent. slip. The *Rapee III*, has an 80-horsepower, 4-cylinder oil engine of 6¾-inch stroke and 960 revolutions per minute. The obtained speed was 21.8 knots with a screw of three blades, 26 inches diameter, 30¾ inches pitch, and about the same slip as above.

I will state here that the weight of an American autolaunch gasoline engine of 25 horsepower, with flywheel, is only 300 to 450 pounds, of the design as shown in Fig. 17, having four cylinders. Even these weights are found to be reduced to ten pounds per horsepower, as I saw on some French makes in St. Louis World's Fair.

A speed of 26 knots was obtained recently with an English craft of only 40 feet length, 5 feet beam and 8 inches draught. The entire weight was 3½ tons, of which about 1.75 tons was for the complete mechanical outfit. There are two gasoline engines, as shown in Fig. 25, each having four cylinders and developing 60 horsepower, making 120 for both. The dimensions are: cylinder, 6½ inches diameter; stroke, 6 inches. The center line of propeller and engine shaft, have a considerable slant, in order to get the blade as deep as possible in the water.

A very small but interesting torpedo boat is shown in Fig. 26. It is of English make and has a speed of 18 knots per hour. The length over all is 40 feet, the beam, 6 feet 4 inches, and the draught 30¾ inches. The capacity is 4½ tons. The engine has been placed forward, as the stern has to be free for the

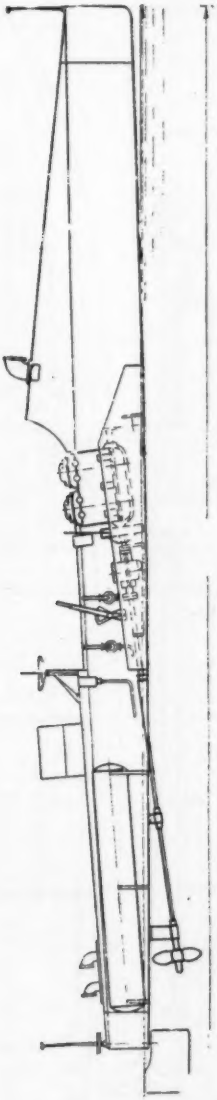

Fig. 25.

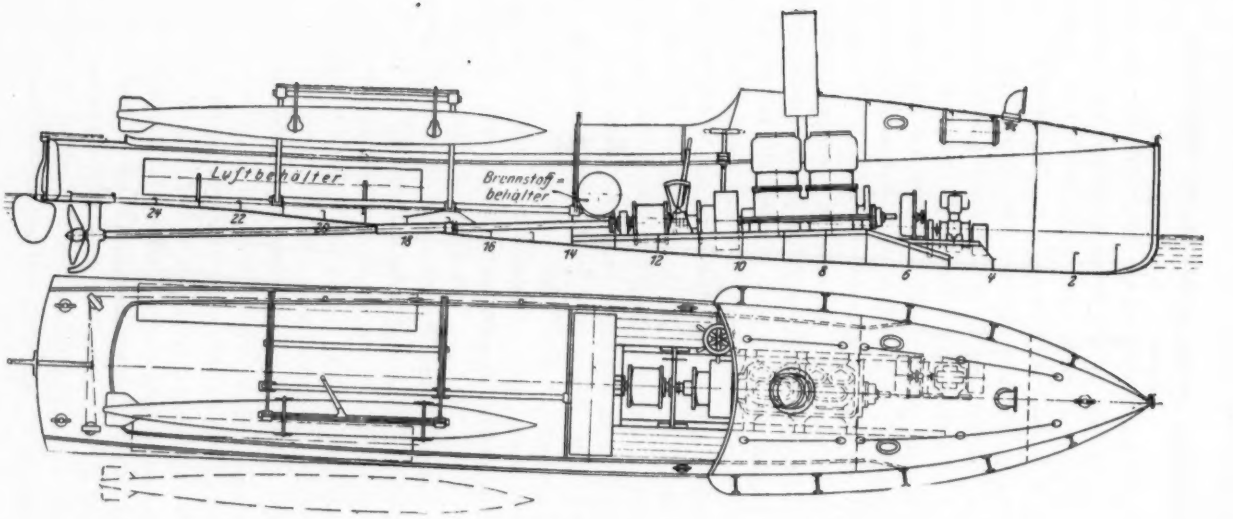
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Fig. 26.

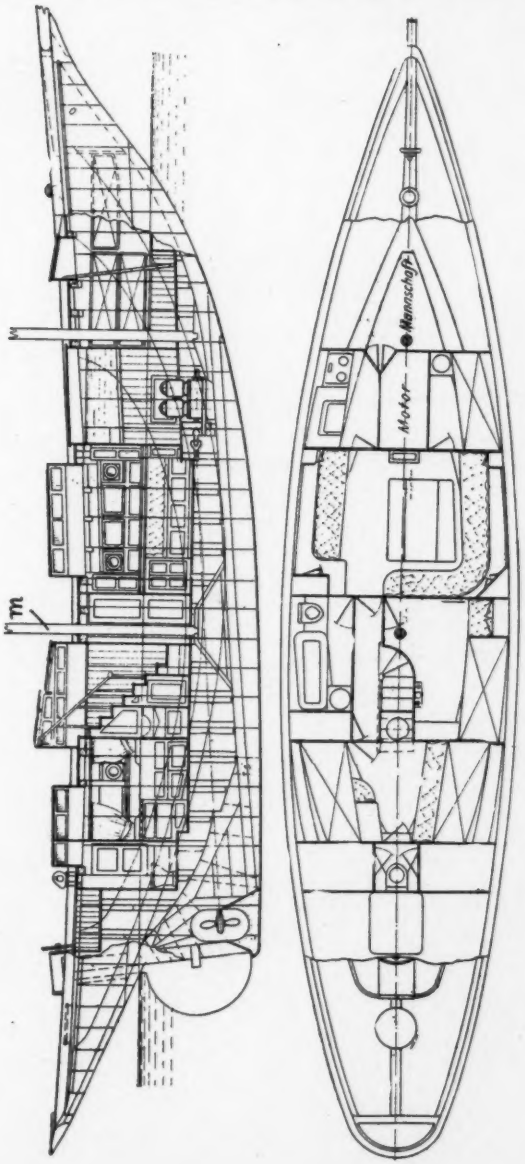

Fig. 27.

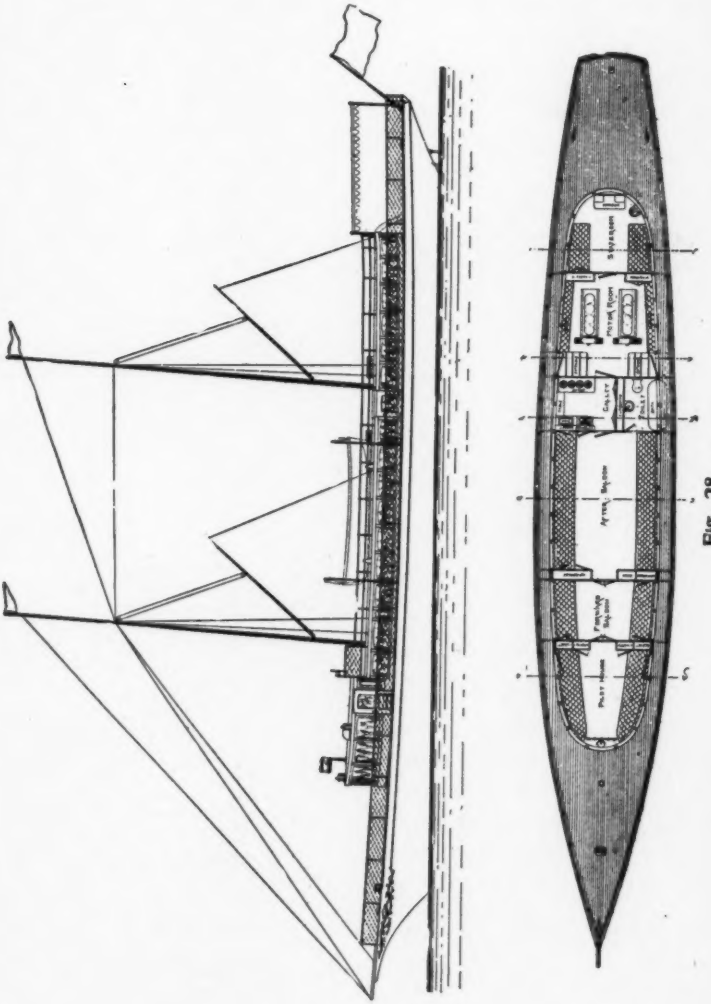

Fig. 28.

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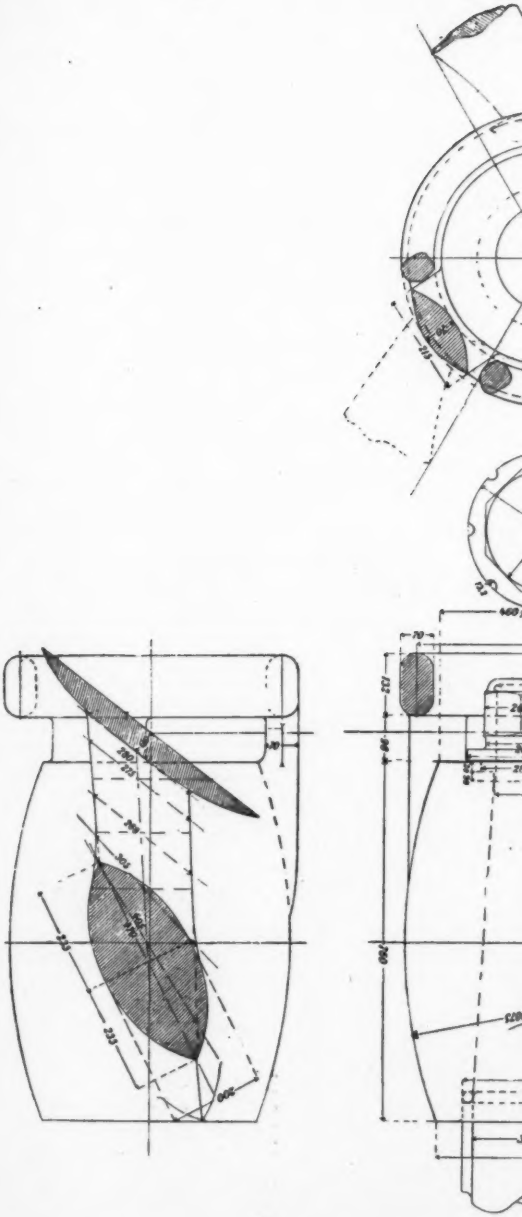


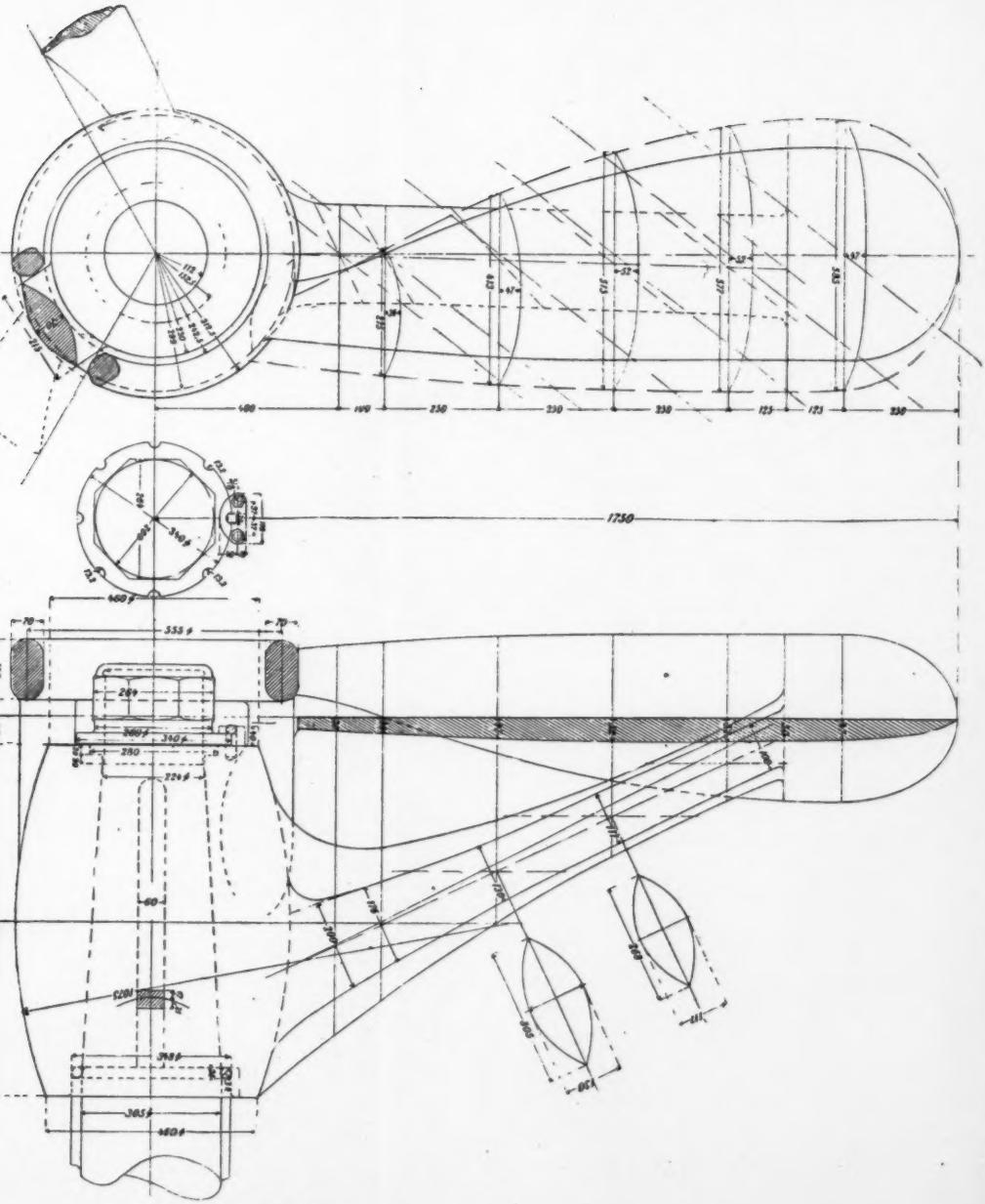


Fig. 29.

torpedo outfit. This engine has four cylinders of 8 inches diameter and 8 inches stroke, and makes 900 revolutions. The rated indicated horsepower is 120. The entire machinery has a weight of 2,750 pounds.

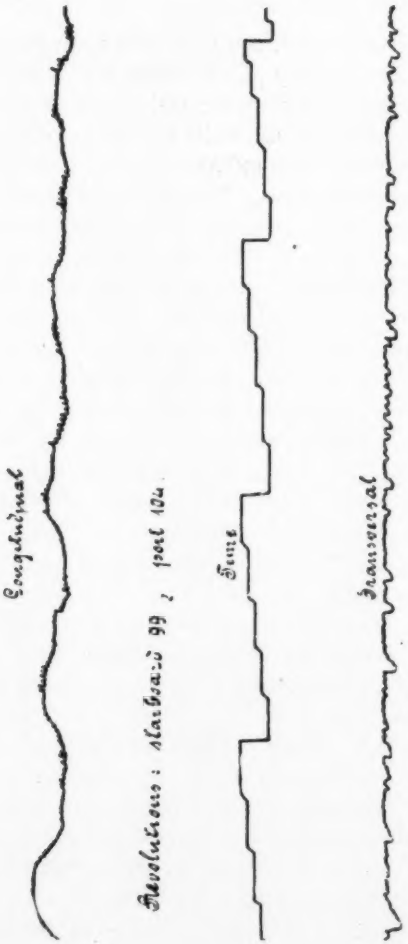
The air for starting is compressed in a separate, vertical, 6-horsepower gasoline engine. Reversing is done by a reversible coupling.

A German pleasure launch equipped with a complete sailing outfit as well as an auxiliary gasoline engine is shown in Fig. 27. The engine has 30 horsepower and runs with 400 revolutions. For fire protection, all walls of the engine room are covered with lead, and a fan exhausts all foul air and gases directly into the atmosphere. The propeller wheel is connected with the engine by means of an universal coupling and the shaft crosses the foot of the main mast M. The speed is 7 knots. The dimensions are: 65 feet length over all and 45 feet on water line; 14 feet beam, and 7 feet draught. The entire area of sails is about 800 square feet.

A 40-foot cabined boat of American make has about 10 feet beam and full accommodations for an extended cruise. The pilot house and saloon are well arranged in front of the after engine room. The boat is generally equipped with a three-cylinder 30-horsepower gasoline engine of about 1,200 pounds weight, and may cost, including the reversing coupling, from $1,000 to $1,200. A 90-foot cruising yacht of fine design and practical equipment, shown in Fig. 28, is of well-known American make. It has about 12 feet beam and 5 feet draught. There are twin screws, each with a reversible coupling connected to a four-cylinder gasoline engine. There is also a complete sailing outfit.

Before closing this chapter it might be interesting to notice one of the recent improvements in propeller-wheel design. For internal-explosion engines with four-cycle combustion, a wheel is particularly desired which reduces the vibrations of shaft and engine. Graf Rudolf Westphalen designed the propeller shown in Fig. 29. This is a wheel for a small cruiser of the German navy. This vessel is 320 feet long, 40 feet

beam, and 15 feet draught, with a displacement of about 2,700 tons. The diameter of the screw is 9 feet 10 inches. The engines develop 8,000 horsepower, and the speed at 160 revolutions is about 21 knots per hour. Figs. 30 and 31 show clearly the difference between the vibrations of an ordinary



Vibrations in Engine-room. Westinghouse Propeller.

Fig. 30.

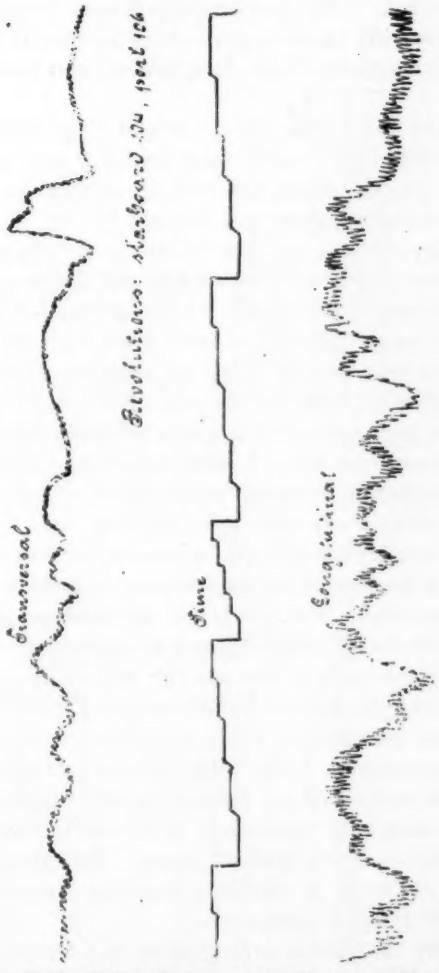


Fig. 31.

screw and the Westphalen propeller, as taken in the engine room with an indicator, the so-called "pallograph" of "Schlick" design. Fig. 31 gives the vibrations in the engine room of the North German Lloyd steamer *Seeadler*, equipped with a propeller of general type, having three blades directly

attached to the hub. Fig. 30 is equipped with Westphalen's propeller, also having three blades. It is of special interest to compare the diagrams of the longitudinal and transversal vibrations.

Now, near the end of my paper, it might be of interest to show some cuts and give some descriptions of coal gas producers of later improvements and designs, as they are in general use for driving stationary gas engines.

The efficiency of a producer plant is about 80 to 84 per cent. Besides the good adaptability of suction-gas plants to 4-cycle engines, and pressure plants for 2-cycle engines, it may be noted that the latter are more suitable for low grades of fuel. The suction-gas producers of today are considered especially adapted to anthracite, coke or charcoal. The efforts of many inventors have, therefore, been to construct suction-gas plants suitable for bituminous coal; I have succeeded in operating one of my own design running on all kinds of soft coal or lignites, even using wood, shavings, corncobs, etc.

All these producers are equipped with vaporizers or economizers for saving the heat of the escaping gas in order to reduce the fuel consumption. This gas is very hot, and one pound of fuel yields about *four* pounds of gas at a temperature of about 1,470 degrees Fahrenheit. The specific heat being taken at 0.23, and allowing 370 degrees Fahrenheit for the gas to enter the scrubber, we have about $1{,}100 \times 4 \times 0.23 = 1{,}012$ B.T.U. for evaporating steam. If the boiler has, in this case, about 80 per cent. efficiency, and one pound of steam requires 1,146.6 B.T.U. latent heat, then this heat is sufficient to make $1{,}012 \times 0.80 : 1{,}146.6 = 0.70$ pound of steam. The steam costs, therefore, do not exist in suction-gas plants, showing their superiority over pressure producers.

Fig. 32 shows the general arrangement of a Deutz suction-gas plant. A is the ashpit, B the producer, D the upper coal magazine, L the chimney valve, and I the gas cleaner or scrubber. K is a gas dryer or purifier.

The wet-scrubber is filled with coke, and water is sprinkled in from the top. The dry-purifier, generally, is filled with

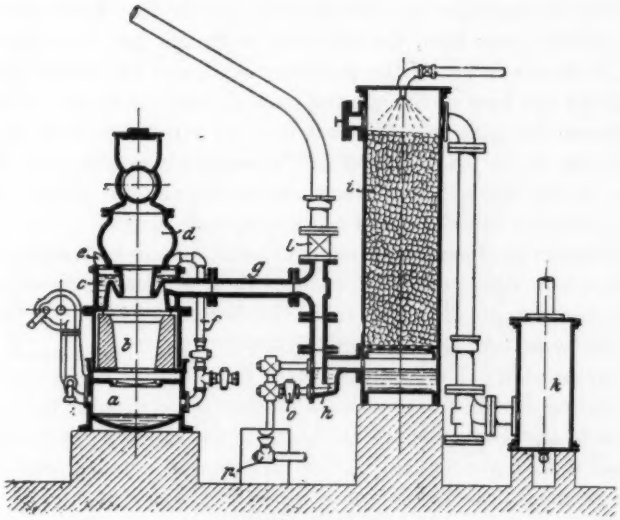


Fig. 32.

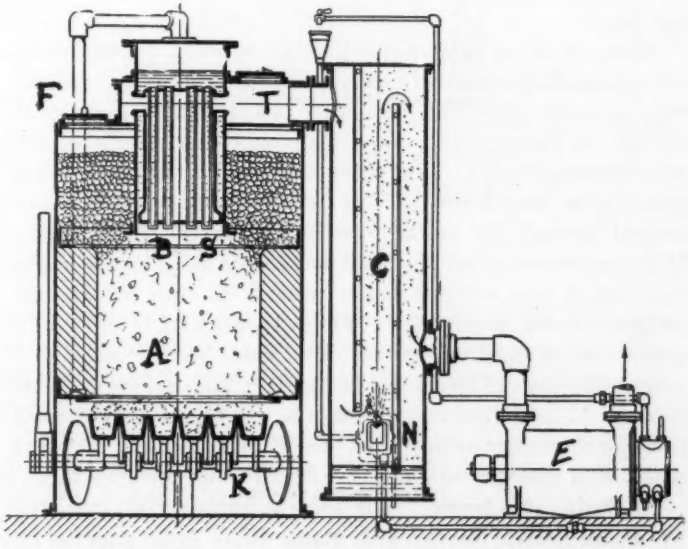


Fig. 33.

sawdust or excelsior in order to take out the fine water particles carried over from the scrubber with the gas. A vaporizer is shown on top of the producer at C, and the steam thus made by the heat of the gas is passed in the lower end of the producer by pipe F. The amount of water entering this vaporizer is, in up-to-date plants, governed according to the load of the engine. In A are flat grates, and I prefer the latter in such cases where a clinkering coal is used.

A special producer for marine purposes, system "Capitaine," is shown in Fig. 33. The producer A is of general construction, but the gas is drawn from the center, very near to the hottest zone, and escapes through T.

The scrubber C is equipped with fine nozzles N, for water sprinkling in the lower end. The coal is fed around the gas pipes S, and center boiler B through the holes F, generally closed with plain covers. There are no stationary grates in this producer, but rocking ones at K, in the ashpit.

The gas, after passing the scrubber, is mechanically cleaned by a tar extractor E of special design, and very high rotating speed.

When it is not desirable to use salt water in producers directly, a special salt-water vaporizer also of "Capitaine" design, Fig. 34, is recommended. In order to avoid the depositing of the salt in the vessel V, there is a continuous current of sea water going through. The percentage of salt, therefore, never goes higher than desired. The fresh water enters at A, passes around through the spiral pipes T, and enters the vaporizer V, in the lower end at B. The overflow is at C, in the upper part, and S is a circular boiler around V, from which the *thickened* water drains at Z. Thus the lost heat is used for a preheating of the fresh water. The water level is kept at a height W, and the steam escapes at P. The hot gases enter through M, and discharge through N, after passing downward through a number of tubes N. I will mention here that it is possible in this way to provide distilled water for drinking.

In closing this paper I will show a foreign boat, operated by an anthracite suction-gas plant. The great superiority

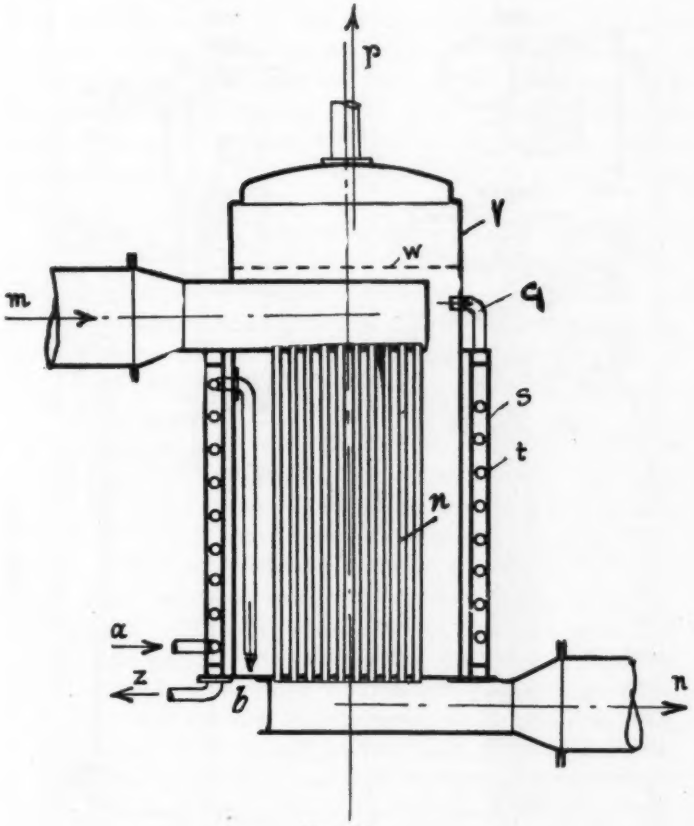


Fig. 34.

over steam outfits is to be found in saving of fuel and men for attendance, and, in comparison with oil engines, in avoiding fire danger by explosions, as well as cheaper fuel costs and saving of bulk.

The first suction-gas boat, *Haldy*, has been running for about 5 years, and the owners are very well satisfied with her. Deutz was soon building more of those boats, and his second

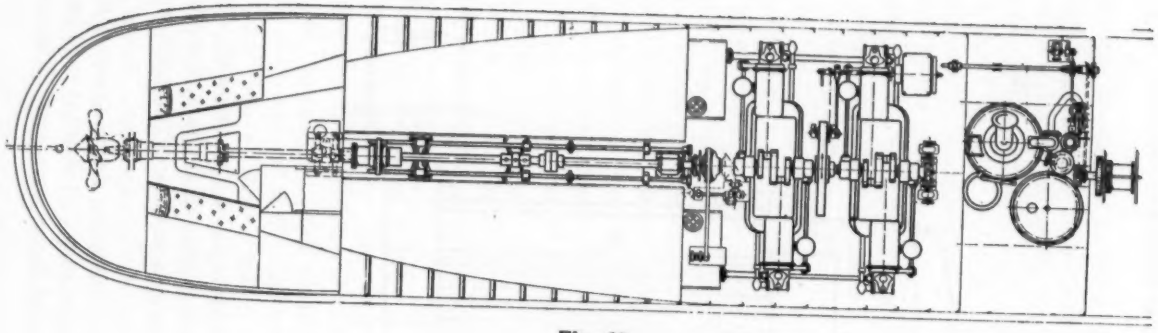


Fig. 35.

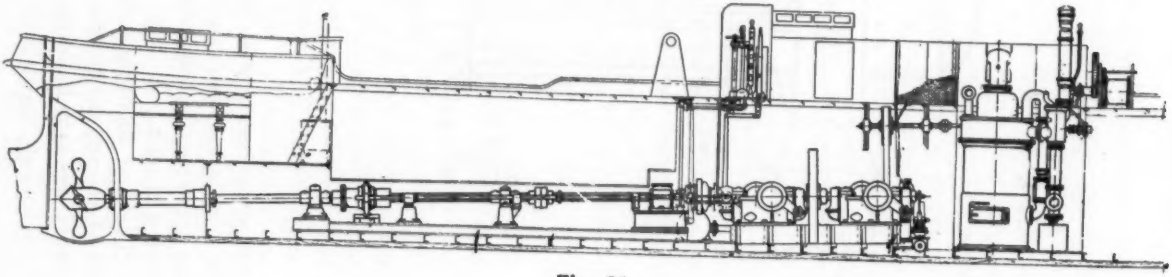


Fig. 36.

boat was the *Lotte*. It has opposite, twin, horizontal gas engines, and one small auxiliary engine for starting.

The *Lotte* is about 131 feet long, 14 feet beam, and of 7 feet draught. The capacity is 240 tons. The horsepower is 100 and the required speed against the current in the Rhine is about $3\frac{1}{3}$ knots per hour. Figs. 35 and 36 give a longitudinal section and a plan view.

The producer plant is between the transverse bulkheads, so that no dust nor gases can enter the other parts of the ship. The propeller wheel is 4 feet in diameter. The auxiliary engine runs on gasoline and develops 6 H.P. It drives a blower and the hoists on deck. Besides, it serves as a starting device for the main engine by means of friction-clutch and belt transmission.

In conclusion I give the following two tables for these boats running on anthracite suction gas. One, Table I, runs regularly 150 knots between Saarbrücken and Mühlhausen, and the other, Table II, runs 167 knots between Cologne and Rotterdam. The boat according to Table I has a 16-H.P. producer plant, 240 tons load, runs 12 days with, and 9 days without, load. It makes 11 trips a year, each 150 knots, and the speed is only a little over one knot, average. The coal consumption is taken with 1.21 pounds per H.P. per hour. The other has a 100-H.P. producing plant, 250 tons load, runs 14 days. The average load up and down is 200 tons. It makes 26 round trips a year and 167 knots for each single route. The coal consumption is taken at $1\frac{1}{3}$ pounds per H.P. per hour. The trip up the river requires 50 hours, and down 25 hours, or 75 hours for the round trip.

A comparison of the expense in operating a freight steamship and freight suction-gas ship shows about 0.16 cents per gas, ton, knot for the latter, and 0.25 cents for the steam, ton, knot. These costs on steam railroads are about five to six times as high as for suction-gas boats.

TABLE I.

OPERATING COSTS FOR SUCTION-GAS BOAT *LOTTA*.

For wear and tear (expenses per year), 5 per cent.,	$200.00
For wear and tear on machinery, 10 per cent.,	250.00
Interest on installation cost, 5 per cent. of $6,500,	325.00
Fire insurance, 2 per cent. of $6,500,	13.00
Taxes and duties,	150.00
Costs for coal for 11 trips, 16 H.P., 12 days with load, 9 days without load, 12 hours, $5.00 per ton, coal,	122.00
Cost for oil, waste, etc.,	61.00
Wages and salaries,	600.00
Total cost, per year,	$1,721.00

This boat makes $11 \times 150 \times 240 = 396{,}000$ knots, tons, per year.

$\frac{1721}{3960} = 0.43$ cents for operating expenses per knot, ton.

TABLE II.

OPERATING COSTS FOR SUCTION-GAS BOAT *HALDY*.

For wear and tear on hull, 5 per cent. of $5,000,	$250.00
For wear and tear on machinery, 10 per cent. of $6,250,	625.00
Interests on installation costs, 5 per cent. of $11,250,	562.00
Fire insurance, 2 per cent.,	22.50
Taxes and duties,	975.00
Costs for coal for 26 trips, 100 H.P., 75 hours, $1\frac{1}{3}$ pounds of coal at $5 per ton,	582.00
Cost for oil, waste, etc.,	244.00
Wages and salaries,	1,750.00
Total for the year,	5,011.00

This boat makes $26 \times 2 \times 167 \times 200 = 1{,}736{,}800$ knots, tons, per year.

$\frac{501.100}{1.736,800} = 0.28$ cents for operating expenses per knot, ton.

I am indebted to the following for much information in preparing this paper:

Truscott Boat Manf. Co., St. Joseph, Michigan; Hercules Gas, Eng. Works, San Francisco, California; Standard Motor Coustruction Co., Jersey City, N. J.; August Mietz, New York; Carl Huber, Berlin, Germany; "Zeitschrift Des Vereines Deutcher Ingenieure," Berlin, Germany; Graf Rudolf Westphalen, Vienna, Austria; Fairbanks Morse Co., Beloit, Wis.

COMPARATIVE TRIALS OF ELEVATING GEARS FOR RAPID-FIRE GUN MOUNTS.

By John F. Meigs.

The comparative trials of elevating gears in which the gun was moved in elevation by means of (1) a single hand-wheel and (2) by two hand-wheels on the same shaft, extended from May 1, 1906, to December 1, 1906, and included the firing of 25,000 rounds. Photograph I shows a Bethlehem 6-inch, rapid-fire gun with which the trials were conducted.

THE APPARATUS.

(*a*) The two-hand elevating gear (Photograph I) consists of two hand-wheels on opposite ends of the same shaft, the handles of the wheels being 180 degrees apart, and the pointer using both hands for controlling the gun. The firing trigger (A) is on the handle of the right-hand wheel and is worked by the index finger of the right hand without releasing the handle.

(*b*) The two-hand elevating gear when driven by the left-hand wheel only, constitutes the usual form of elevating gear. In this case the firing was done by means of the ordinary pistol grip (B). A second pointer pointed the gun in azimuth from the right-hand side in both cases.

(*c*) Three sets of speed gears were used by each pointer at each range, and for one complete turn of the elevating hand-wheel moved the gun as follows:

No. 1 Gear 30 minutes of arc.
No. 2 Gear 50 minutes of arc.
No. 3 Gear 75 minutes of arc.

(*d*) A sub-caliber rifle, shown in Photographs III and IV, was rigidly attached to the muzzle of the gun and fired electrically by the firing gear of the big gun.

(*e*) The targets used represented a target twenty-six feet high at the different ranges, and were actually 1.37 inches high for 1,500-yard range; .68 inch high for the 3,000-yard range; .34 inch high for the 6,000-yard range, and .23 inch high for the 9,000-yard range.

The target was moved to simulate the continuous rolling of a ship from 1 to 10 degrees (Photographs III and IV) by means of a cord, one end of which was fastened to the target, and the other end, after passing over a pulley, was moved continuously over the semi-circumferences shown in Photograph II.

(*f*) The curve by which the target was moved up and down simulating the rolling of a ship (Photograph II) is made up of semi-circumferences joined together, and the radii of these are of such lengths as to cause the target, and consequently the gun, to move through from 0 degrees to 1 to 10 degrees depression or elevation respectively, and back to 0 degrees again. The time allowed for this (7 seconds) was the same for each roll, and corresponded to a 14-seconds period for the complete oscillation of a ship. The semi-circumferences were sub-divided into two-second periods, and a time-bell enabled the operator to pass each mark at the proper time.

GUN POINTERS.

Eight gun pointers were in competition throughout the trials, and while the final scores differed slightly, the relative percentages given by the table were practically obtained by each pointer.

CONDITIONS.

(*a*) Each pointer was allowed a trial shot before each string for adjusting the sight. The distance from the sight telescope to the target was 20 feet.

(*b*) Each pointer fired for five minutes continuously with a six-seconds loading interval after each shot.

TABLE OF RECORDS.

The total time of firing as between the records for one-hand and two-hands are equal. For example, the time of firing with one and two hands at the 1,500-yard target and on the rolls 10 degrees to 8 degrees are the same.

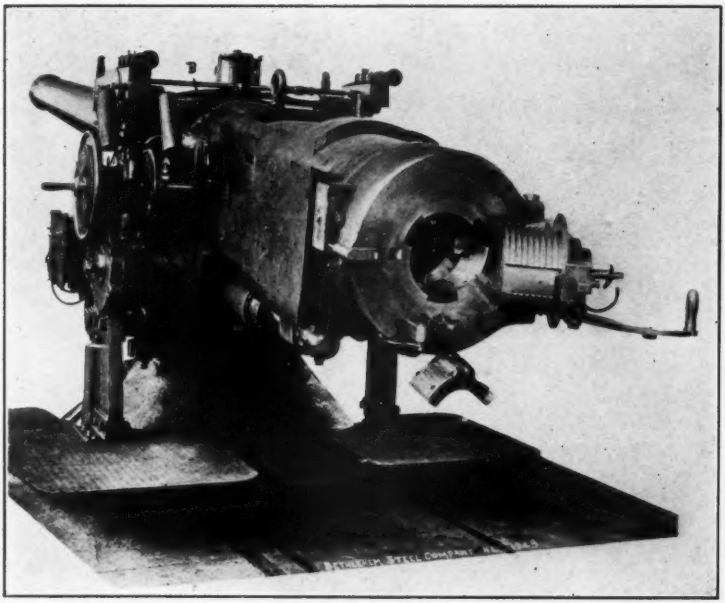
.....	Range 1,500 yards.			Range 3,000 yards.			Range 6,000 yards.			Range 9,000 yards.		
	Groups of rolls.			Groups of rolls.			Groups of rolls.			Groups of rolls.		
	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°
Total shots, 1 hand..	217	356	580	442	1328	1702	275	1055	1377	240	856	1481
2 hands.	242	403	642	518	1544	1837	332	1232	1467	290	1042	1589
Total hits, 1 hand....	153	285	518	225	777	1376	81	442	803	61	286	757
2 hands...	188	345	582	334	1055	1580	128	553	924	101	437	969
Gain in speed of hitting, 2 hands over 1, in per cents.....	22.9	21.1	12.3	48.4	35.8	14.9	58.0	25.1	15.1	66.6	52.8	28.0
Gain in accuracy, 2 hands over 1, in per cents.....	10.2	6.9	1.6	26.7	16.8	6.4	30.5	2.5	5.2	37.0	25.5	19.4

NOTES ON TABLE.

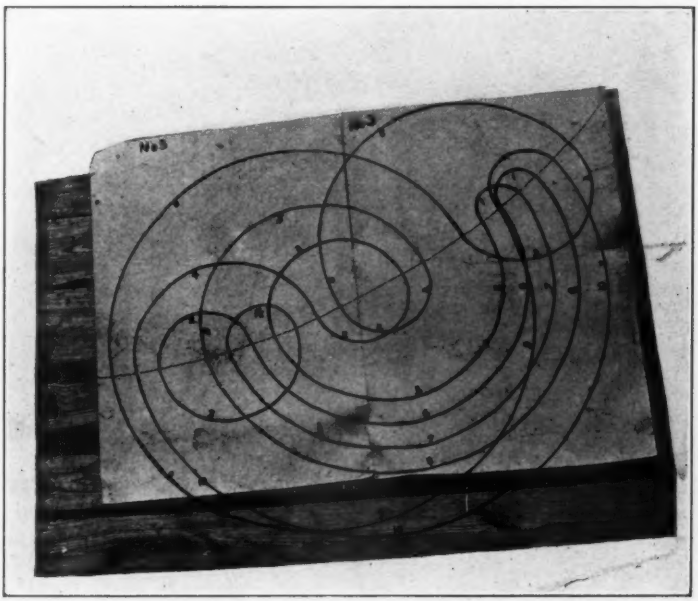
For an actual example, which might occur in practice, we take the figures above at 3,000 yards range and with the ship rolling from 8 degrees to 10 degrees. Imagine two 6-inch guns side by side, one with a one-handed and the other with a two-handed drive. Suppose these guns fire for the same time, with pointers of the same skill, and at the same target. If 100 represents the number of hits made with the one-handed gun, then 148.4 will represent the hits with the two-handed one. Also, under the same circumstances of firing, the percentages of accuracy show that in expending 100 rounds from a magazine, firing at the same speed as above, we shall get 50.9 hits with the one-handed gun, and 64.5 with the two-handed gun. The other shots will be misses. This latter statement is equivalent to this: when firing at the highest speed convenient to pointers, the expectation of getting a hit with any one shot, in the circumstances defined is, with the one-handed gun $\frac{225}{442}$, and with the two-handed gun $\frac{334}{518}$. These fractions are the same as .509 and .645.

Also, the figures show an increased rate of hitting with two hands (an increase in the number of hits per minute) under all circumstances from the greatest to the least difficult. When firing at the nearest target (1,500 yards) and on the rolls 1 degree to 3 degrees, the increase in the rate of hitting is 12.3 per cent., or for every 100 hits in a given time with the one-handed drive there will be 112.3 hits with the two-handed drive. When firing at the most distant target (9,000 yards) and on the rolls 8 degrees to 10 degrees, the increase in the rate of hitting is 66.6 per cent., or for every 100 hits in a given time with the one-handed drive there will be 166.6 hits with the two-handed drive. As stated, in all cases where the difficulty of the practice is intermediate between these two extremes, the two-handed drive shows more hits per minute than the one-handed drive.

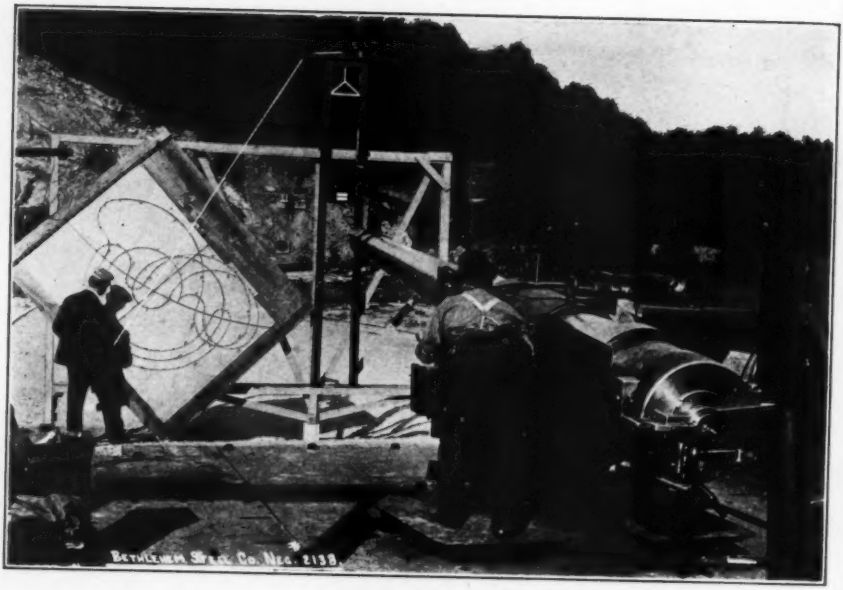
The accuracy of the practice, independently of the firing, is greater in all cases with the two-handed gear. Taking the



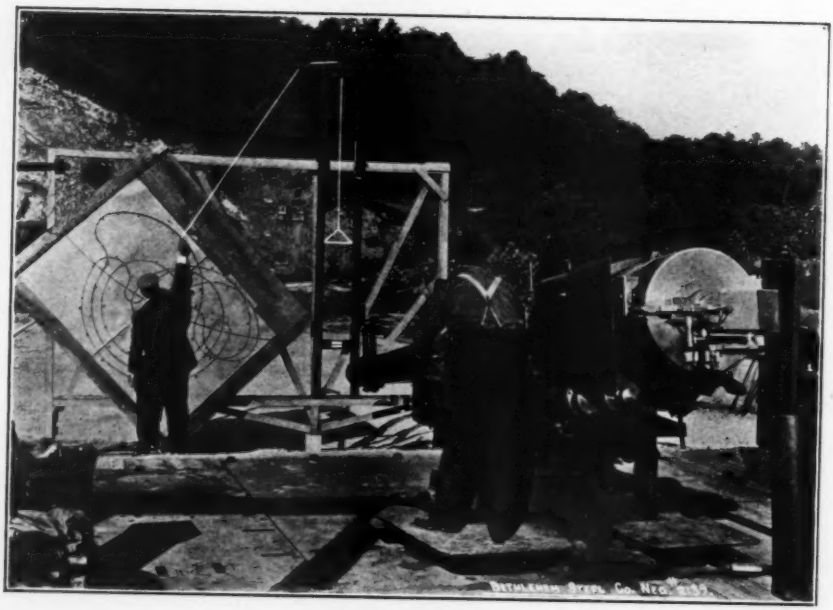
Photograph I.



Photograph II.



Photograph III.



Photograph IV.

two extremes of difficulty as before, the record shows that with the 1,500-yard range and the rolls 1 degree to 3 degrees, the expectation of hitting with any one shot is, with the one-handed gear, $\frac{518}{590}$, or .893, and with the two-handed gear, $\frac{582}{642}$, or .907. With the 9,000-yard range and rolls 8 degrees to 10 degrees, the expectation of hitting with any one shot is, with the one-handed gear, $\frac{61}{240}$, or .254, and with the two-handed gear, $\frac{101}{290}$, or .348. In considering these last figures it must be remembered that they show only the increase in the probability of hitting with any one shot. The increase in the rate of hitting per minute is greater than these figures, and is as already given above. This is because gun pointers will always fire more shots per minute with the two-handed drive than with the one-handed drive. They get their sight on the target oftener. For example, taking again the circumstances of least and greatest difficulty, the record shows that with the 1,500-yard range and in rolls 1 degree to 3 degrees, the number of shots fired with the two-handed gear will be as 642 to 580, as compared with the one-handed gear; or, the two-hand gear gains in speed of firing, $\frac{62}{580}$, or .11. At the 9,000-yard range, and with the rolls 8 degrees to 10 degrees, the number of shots fired with the two-handed gear will be as 290 to 240, as compared with the one-handed gear; or, the two-handed gear gains in speed of firing, $\frac{50}{240}$, or .21. The table of records shows that there is no interruption in the laws of accuracy and speed here shown.

It is to be noted that the results as to accuracy and speed of firing here shown cannot be reached in actual practice with targets at a distance. As the targets were only four feet from the muzzle of the small gun used, the bullets went straight to them, and all inaccuracies due to imperfections in powder and knowledge of range are eliminated. The records show what might be attained if muzzle velocities were very much greater than they are and were maintained, and measure therefore only the accuracy attainable by pointers with service sight and service arrangements for handling the gun.

GENERAL CONCLUSIONS.

The better results obtained by the two-hand elevating gear as shown by the table are accounted for by the following conditions.

(1) The gun can be worked faster with two hands than with one, and the larger the gun the greater the advantages.

(2) The gun can be kept on the target better on account of the ease with which it may be stopped, started or reversed.

(3) The motion of the gun is smoother on account of the greater power to overcome tight places in the gears and also the absence of a dead center with the two-hand drive.

(4) The motion of the gun pointer's body is less with the two-hand drive, and the eye can be kept constantly at the telescope.

DIFFICULTIES EXPERIENCED WITH THE MAIN ENGINE BEARINGS ON SOME OF OUR LATEST VESSELS.

By Lieutenant A. F. H. Yates, U. S. N., Member.

Owing to the wide amount of attention attracted of late by many instances of failures of the main engine bearings of our new vessels, the writer has collected certain data in the case of five vessels, which it is hoped will be of interest to the readers of the Journal.

These failures have resulted in much discussion as to the exact cause, it being believed by some that they were due to the quality of the anti-attrition metal used, and others attributing them to the use of a poor quality of lubricating oil. For this reason special effort has been made to bring out all matters in these connections.

To begin with, the anti-attrition metal required for installation by the builders in all of these vessels is known as the Bureau of Steam Engineering Standard, its composition being as follows :

Best refined copper, per cent., . . .	3·7
Banca tin, per cent., ...	88.8
Regulus of antimony, per cent., . . .	7·5

by weight, to be well fluxed with borax and rosin in mixing.

To avoid the advertisement of the various oils used, the brands are not mentioned, but it should be remembered that during all of the official trials of these five ships, previous to commissioning, only a well-proved brand of mineral lubricating oil was used, or else lard oil. It was only after commissioning that the so-called specification oils were used.

"BATTLESHIP A."

Preliminary Trial.—During the four-hour run of the preliminary trial of this vessel the main engines worked well except for a hot valve-rod guide on the starboard engine, necessitating slowing down, and the port intermediate guide, which ran warm continually. The trial was successful and the engines were finally reported as entirely satisfactory. A special marine-engine oil was used by the contractors.

The port H.P. crank pin ran hot at a later date, while making 115 revolutions on the measured mile at Rockland, Me., due to a poor quality of lubricating oil used.

Final Trial.—The four-hours' full-power, forced-draft trial was abandoned after the vessel had run an hour and three-quarters, on account of the heating of bearings No. 1 and No. 2 of the port crank shaft and the running of white metal in same. The port crank shaft worked forward from the thrust block until the forward webs of the port forward L.P. and port H.P. and port I.P. crank shafts took on the after fillet of main bearings No. 1, No. 2 and No. 4, melting the babbit metal on the fillets of these brasses. The starboard H.P. crank-pin brasses, the guides of the port L.P. crosshead, and those of the port after L.P. valves also ran hot, although water and oil were used profusely on them. The hot bearings were cooled and the engines ran smoothly for about one hour at the high speeds, after which time the bearings began to run warm, and, although oil was profusely used on them, it was not possible to keep them from heating. Even those bearings which had been very carefully adjusted did not seem to run more satisfactorily than the others.

Owing to the heating of so many bearings, the running of the white metal in some, and the heavy pounding in the starboard H.P. crank-pin brasses, caused by their having been overheated and flattened, it was decided not to run the engines at their maximum speed, and they were therefore run at a moderate speed of approximately 85 revolutions (about 13 knots). The cause of this failure was the use of a poor grade of oil (specification oil).

This trial having been unsatisfactory, it was recommended that *Battleship A* be given another final acceptance trial as soon as a proper adjustment of brasses could be made, using the same high grade of lubricating oil that was used successfully by the contractors on the preliminary trial. The oil used by the builders on the preliminary trial was the same brand as was used by the builders of a sister ship on her trial, which passed off most successfully without the use of any water on the bearings.

It was further recommended that steps be taken to locate the cause of the trouble experienced with *Battleship A's* bearings, whether due to faulty design, faulty babbit metal, or poor quality of lubricating oil.

The vessel returned to a navy yard to have the necessary adjustments made, and during her stay, an investigation was made. Upon examination of main bearings No. 1, No. 2, No. 4 and No. 5 of the port engine, it was found that the after ends of the babbitting of each had softened and pulled off as a result of heating. It was decided to trim out the ends and freeze on other babbit metal. The brasses were then refitted to the journals and the fillets rounded off. The forward and the after L.P. crank-pin brasses of the port engine were found to have too limited clearance (a scant $\frac{1}{16}$ inch of the forward L.P. and a full $\frac{1}{32}$ inch of the after L.P.), at the forward ends. This clearance was increased by cutting off $\frac{1}{16}$ inch from the forward face of the brasses in each case. An examination of the starboard crank-pin brasses showed that the babbit metal had dragged and had completely filled up the oil holes and oil grooves. The metal had also burned. The ship being under hurry orders to be ready for her trial runs, the spare brasses were fitted and the ones removed were re-babbitted. Upon removing the crank-pin brasses of the port H.P., the babbit metal was found to have dragged to some extent and to have partly filled the oil ways, as with the starboard H.P. brasses. There were also cracks in the white metal. The urgency attending the getting away of the vessel prevented anything more being done than to refit the brasses,

but it was recommended that these brasses be re-babbitted and refitted when the vessel should return from sea. Prior to the vessel going to the yard, the port thrust bearing had been set forward $\frac{1}{16}$ inch to increase the crank brasses' clearances, and it was decided to readjust them to the original position, in order that the crank shaft could be moved aft $\frac{1}{16}$ inch.

Repairs having been completed, the vessel proceeded to sea for her trials.

Three hours after the four-hours', full-power, forced-draft trial was started, the port main thrust collars, forward end, ran warm, but were kept under control. All machinery worked well until one-half hour later, when the forward end of the port thrust warmed up again. The forward H.P. crank web began striking on the after end of the cap to No. 2 main bearing on three of the steel parts of the fillet, emitting sparks at each of these places, and at the same time the port H.P. crank pin began warming up on the inboard side; the outboard ahead side had already heated, as was afterwards shown. Water was put on each of these places and an effort made to keep them cool until the end of the run. Several minutes later the babbit in the port H.P. crank pin ran, and, the resultant knock in that bearing being dangerous, the engine was stopped as quickly as possible, and the trial was stopped. The starboard engine was slowed to 40 revolutions and the port engine and its auxiliaries secured and the jacking gear thrown in. The work of taking the port H.P. crank pin and No. 2 port main bearing apart was promptly undertaken. The main bearing was found to be in excellent shape, and the after end was filed smooth where it had taken on the web of the crank, and replaced at a 28 B.W.G. lead. The port H.P. spare brasses were scraped in and fitted as well as possible, though the forward end of the brasses came fairly hard up against the forward web of the crank. The brasses were adjusted to a 26 B.W.G. lead on the bottom and a 28 B.W.G. on the sides. This work was finished the next day, and the engines were warmed up and turned over, and an hour later run up to a speed of 105 revolutions, when the port H.P.

crank pin began running warm again and the trial was given up. An investigation of the port H.P. crank pin and brasses showed that the babbit on the outboard side where it was very thin had worn away until the pin had come in contact with the brass, the brass and pin had expanded, the pin was cut, and the forward edge of the web took on the fillet of No. 2 main bearing cap.

The port H.P. crank pin had been refitted to a 28 B.W.G. lead just prior to the trial, but, as the brass had not been properly centered between the bolt holes when roughed out prior to babbitting, the babbit was too heavy on one side and too light on the other. There was also a slight crack in the babbit in the bottom near the crown, and as the time was too short to re-babbit the brasses it was decided that the brasses would run the trial satisfactorily, but that they should be re-babbitted upon its conclusion.

Several bearings that heated on the previous trial, when using specification oil, ran cool during this second trial, when using a selected oil. This would indicate that the quality of oil was in some measure responsible for the heating of the bearings during the previous trial.

Finally, although the brasses had been carefully adjusted at the yard, the vessel failed to complete her second, four-hours', full-power, forced-draft, final trial, owing to the melting of babbit metal in the port H.P. crank-pin brasses. In view of the fact that a first-class grade of lubricating oil was used throughout this second trial, it was concluded that the failure of this trial was due to the babbitting and partly to improper adjustment. The crank-pin and main bearing brasses were undoubtedly not properly bored or babbitted in places, especially the starboard H.P., port H.P., and port forward L.P. crankpins. Their spares were in the same condition.

The following are some other conclusions that were expressed in connection with this failure :

(a) The oil was good and did its work well, as shown by the condition of No. 2 main bearing after the trial.

(b) The oiling service is inadequate for steaming at full

power. An efficient centrifugal oiler is necessary for the crank-pin brasses, and larger oil pipes and automatic oil gear should be installed. The oil gear for the eccentrics could be much simplified by telescopic oil gear.

(c) The centrifugal oiler suggested should be fitted so that the bearings could be flooded with either water service or from buckets filled with soap and water or oil. As now arranged, when the oil grooves are filled and the bearing begins to pound or run warm, there is no way of getting either oil or water to the heated part. The water service merely drops water on the ends of the bearing, and very little if any penetrates to the center of it.

(d) More clearance should be left between crank-pin brasses and crank webs, and between crank webs and main bearing brasses on the forward end of the web.

(e) Greater margin of bearing surface should be given on all crosshead, crank-pin and main bearings.

(f) The design of the brass, with all the babbit in one slab, is poor, as any knock or heating causes the metal to flow in the direction of least resistance. On several occasions metal had to be chipped off the ends of the brasses and out of the oil grooves, when bearings heated while running at slow speeds, bearings set to a 30 B.W.G. lead.

It might be remarked here that, owing to the limited amount of time available for repairs, none of the changes last mentioned were made, though a change in the oiling gear for the eccentrics, to arrange the run of the oil piping with a continuous descent from cup to strap, instead of running for a considerable distance in a horizontal direction, was authorized.

The fault in making the brasses and in babbitting consisted in the axis center between bolt holes in brass castings not coinciding with the center line of the cored-out hollow for babbitting, causing the babbit to be heavy on one side and thin on the other, at one end, and just the reverse on the other. The holding pieces on the brass casting should have been recessed also, instead of filletted, as they were in places,

so that the babbit, when being hammered in, would hold instead of riding up over those pieces, and becoming loose.

"*BATTLESHIP B.*"

Preliminary Trial.—The working of both the main and the auxiliary machinery during the preliminary official trial of this vessel was most satisfactory. As usual on preliminary trials, a large amount of lubricating oil was used, about 540 gallons being expended during the four-hours', full-power, forced-draft trial. A special marine-engine oil was used throughout the trial.

A chemical analysis of the "specification" oil used on this vessel was made at a navy yard several months after she had been placed in service, as it had not given entire satisfaction. Two samples were furnished, one from the top of a tank in starboard engine room and one from the bottom of the tank.

RESULTS:

	Top of Tank.	Bottom of Tank
Specific gravity at 60 degrees Fahrenheit	.9237	.924
Flashing point, degrees Fahrenheit	407.	405.
Free fatty acid, as oleic, per cent	.70	.70
Viscosity (Engler) at 90 degrees Fahrenheit	705.	707.
Viscosity (Engler) at 150 degrees Fahrenheit	163.	164.

These two samples were said to have been taken from a a tank which had remained intact since first filled about two months previous.

The tests show them to be practically identical, no separation of the ingredient oils having taken place.

With these samples were submitted small residues said to have been obtained by straining one gallon of each of the above oils. That from the top oil, amounting to about 1 part in 30,000 of the oil, consisted of siliceous and scaly dirt, evidently from the tank, which had not been thoroughly cleaned. That from the bottom oil, amounting to about 2 parts in 30,000, contained a small amount of dirt, but consisted mainly of metallic oleates (principally of zinc) formed by the action of the fatty acid on the inside coating of the tank. Traces of iron and copper were also present.

In addition to the above samples a sample from a tank in the port engine room, the greater part of the contents of which had been drawn off and strained, was compared with a sample of the strained oil. These were found to be identical in specific gravity and viscosity. The residual oil contained a small amount of the soft, white zinc oleate mentioned above.

Another residue obtained by straining consisted mainly of tobacco, scrap and cotton waste.

The senior engineer officer stated that the oil flowed freely and showed no gumming properties, these results confirming the results of previous tests made in the laboratory when the oil was first received at the Yard. The samples taken for test at that time were perfectly clean and free from dirt.

Final Trial.—A speed trial was made under full power, natural draft, as a portion of the acceptance trial of the ship. The engines were run at an average speed of 105 revolutions for eight hours, at which speed the crank-pin brasses of both I.P. engines pounded siightly and the crosshead brasses of both H.P. engines pounded badly—so badly that it was deemed advisable to heave to for three hours to adjust them before proceeding with the remaining trials at higher speeds. On the conclusion of this eight-hour run a twenty-four-hour run, at 75 per cent. of the I.H.P. developed on preliminary full-speed trial, was begun. The engines were run at about 112 revolutions for three hours. The thump in the I.P. crank pins previously noted grew steadily worse, and at the end of the third hour the thump in the I.P. starboard crank pin became so heavy that the brass heated from percussion and the white metal melted and ran out of the bearing; the trial was, perforce, discontinued and the ship proceeded into port at 11 knots. Before proceeding to the Navy Yard a spare set of I.P. crank-pin brasses, furnished by the builders, was fitted to the starboard I.P. pin; the old brasses were found to be in such condition that it was not deemed safe to try to run up the harbor with them; the soft-metal lining of the brasses was mashed and pounded down badly. The crank pin was scored and rough; it was smoothed before the brasses were fitted.

On the run to the Navy Yard the new I.P. crank pin brasses ran warm. On examination, after arrival at the Yard, it was found that these new brasses, after only one hour of running, were pounded and mashed down badly, the metal filling all the oil grooves. The conclusion was that the metal was entirely too soft.

All the crank-pin brasses were refitted and all were babbitted, except the after starboard L.P. and the two port L.P. brasses. Nearly all of the brasses were found to be sprung out of shape, the back of the brass, in each case, being high in the center over the crown of the brass; also, the bolt holes in the brasses were not in line. The concensus of opinion was that the brasses were neither strong enough nor thick enough at the crown.

During the first run made by the vessel after leaving the Navy Yard the two H.P. crank pins pounded slightly. The brasses were examined and overhauled, and the metal of the starboard H.P. was found to be mashed down into the oil grooves. The starboard I.P. brass, which had been scraped and fitted to the pin at the Navy Yard, was also examined and was found to bear only on the sides and not at all on the crown, as if the brass had again sprung at the center. These brasses were then scraped and fitted to the pins and set up to No. 29 B.W.G.

On getting under way for a run of 90 miles, both engines were found to run well, the port engine especially. At first there was a slight thump in the starboard H.P. crank pin; this thump steadily increased, and at 4 P. M. was very heavy —so heavy that the overhauling of the brass was deemed absolutely necessary before making a long run at sea. On examination, the metal of the starboard H.P. crank-pin brass was found to be spread and mashed out into the oil grooves, and on the cheek-face of the brass, where, at starting, the ends of the white metal had been flush with the surface of the brass, the white metal had been pounded out until it projected $\frac{1}{64}$ inch beyond the ends of the brass, and this after only 90 miles of running.

On a five-day run at sea the starboard H.P. crank pins and crosshead pins pounded heavily; on examination, the metal of the starboard H.P. crosshead brasses was found to be mashed slightly, the oil hole in the cup brasses being covered. The port H.P. and port I.P. crank-pin brasses were examined the next day and were found to be brass-bound, while the oil grooves were partly filled with mashed out metal.

This led to a recommendation that the character and qualities of the white metal installed be investigated, especially with reference to its ability to stand the heavy work imposed upon it at high speeds; also that investigation be made to determine whether or not the crank-pin brasses were sufficiently thick through the crown to maintain their shape under moderate pounding. It was ascertained that the same mixture of babbit metal was used for the bearings of *Battleship B* as was used for four other ships by the same builders. This metal was received by the builders from the sub-contractor in a lot, and was similar to the mixture employed by the builders.

In the case of a number of the crank-pin brasses, some of which were re-babbitted at a navy yard, it was noticed that the metal mashed out after a short while at the ends of the brasses so that the white metal projected beyond the surface of the fillet at the ends of the brasses.

"*ARMORED CRUISER C.*"

Preliminary Trial.—About one hour and a-half after the four-hour, full-power, forced-draft trial had started, the starboard engine was slowed down and almost stopped, on account of a knock in the forward L.P. crank-pin brasses. An examination showed everything apparently all right, but when the engine was started up again a knock was heard, and the engine was stopped immediately. The inboard distance piece was found lying in the crank pit, the dowel pins having sheared. During the second four-hour run the fourth main bearing of the starboard engine ran hot, when a full force of water was turned on for half an hour and the bearing was

slacked off. An oiler subsequently found a piece of metal in this bearing. After this no trouble occurred. The contractors used a high grade of lard oil on this trial.

Several months after the preliminary trial was held it was reported necessary to examine and adjust all crosshead, crank-pin and main bearings, most of them appearing to need re-babbitting. All of these bearings on both H.P. and I.P. engines gave trouble continually during a run of about 4,500 miles, every available minute in port during the trip being employed by the ship's force in overhauling them. The port I.P. crank-pin bearing was replaced with a spare one. It first ran warm, and at the end of the return run was knocking badly. The metal appeared to be too soft, having dragged around in flakes, completely filling the oil grooves and accumulating in bullets and rolls inside of the distance pieces. The three crank-pin brasses examined showed the white metal drawn down from the edges of the brasses more than half way around. The lower brasses of the main bearings were not examined but were judged to be in as bad condition as the caps. Owing to these conditions a board of officers made an examination of the bearings. All the bearings of the main engine that were open to inspection were examined, and the white metal was found to be in such bad condition that it was reported necessary to re-line the crank-pin and crosshead brasses, examine when practicable the go-ahead eccentrics and refit the main-bearing brasses throughout. The bearing metal showed such excessive wear, and in many cases such a loose condition, as to make the above work necessary to place the machinery in efficient condition for service. The metal surfaces were found to be dragged in irregular patches, completely obliterating the oil grooves, and the remaining metal in such condition that re-boring would provide too small an amount for wear. It was not possible to examine all of the bearings, owing to the magnitude of the work of stripping them, by the force on board, but from those examined it was apparent that all were in equally bad condition. From the appearance of the bearings and their general history it was

concluded that the primary cause was the use of an inferior quality of oil.

While there was no record of the bearings having run excessively hot, it was stated that it was impossible to run them cool, even with an excessive supply of this oil and at comparatively low speeds, without using water service.

A chemical analysis of the babbit metal used in the bearings of *Armored Cruiser C* was made at a navy yard with the following result:

Copper, per cent.,	4.30
Tin, per cent.,	88.05
Antimony, per cent.,	7.53
Lead, etc.,	Traces.

This corresponds closely with the mixture prescribed.

The oil used on *Armored Cruiser C*, as stated above, was specification oil. It was received on board in a lot of 2,000 gallons. This oil had been rejected on first inspection, but was finally passed, and 1,941 gallons of the oil was used. On account of indications of hot bearings orders were issued to be liberal in the use of oil. During the vessel's run of about 4,500 miles, the four H.P. and I.P. crank-pin bearings ran warm five times, and three main bearings once each. As a rule the warm crank-pin bearings followed close upon readjustments.

Sample bottles of the oil taken from the gauge glasses shortly after receipt on board showed from $\frac{1}{8}$ inch to $\frac{1}{4}$ inch fairly solid sediment. Samples taken from a 1,000-gallon tank with only 150 gallons remaining showed about the same. Eleven gallons of sediment were taken from a tank that had contained 1,100 gallons of oil. Of this two gallons were of the consistency of soft soap.

Water was kept constantly on the H.P. and I.P. crank-pin brasses, first to reduce the temperature and afterwards to reduce the knocking. The oil did not produce a lather with either fresh or salt water, and did not appear to cling to the bearings. This was noticeable on the main bearings where

the shaft was nearly dry on the inboard side, even when oil was being poured into the outboard side of the hand hole.

Armored Cruiser C was not in readiness for her final trial on the contract date, the re-babbitting of her bearings not having been finished.

As it was thought possible that the failure of the crank-pin bearings of this vessel was, in part, due to an insufficient supply and bad distribution of oil, two more oil holes in each crank-pin bearing were drilled, near the ends, with tubes and oil boxes located alongside the present ones, using drip-feed boxes, without sight-feed attachments.

Subsequent to the above an examination of samples of the oil used, of the bearings that had been removed, of pieces of solid metal from the bearings, and of specimens flaked off, was made by the builders of the vessel. The analysis of the solid metal was as near as possible that which was originally put in (Government mixture), while that of flaked specimens was very much different, being mostly tin, which melts at a very low temperature. The analysis made showed the following compositions:

	Tin.	Antimony.	Copper.	Lead.
Government standard.....	88.8	7.5	3.7	...
Port H.P. crank pin.....	89.57	6.66	3.7	.07
Flake from port H.P. crank pin.....	94.8	3.95	1.12	.14

This examination would seem to indicate that the metal of the bearing was, in spots, heated to above the melting point of tin (446 degrees), when the tin separated from the other metals and "flaked" or "dragged." The heating might have occurred from bad oil, too snug fit, or from an improperly-fitted bearing; most probably the causes in this case were bad oil and too snug a fit.

"*ARMORED CRUISER D.*"

Preliminary Trial.—The four-hour official trial of *Armored Cruiser D* was accomplished without the least trouble arising to cause a pause for an instant. The main engines ran with remarkably little vibration, none of the main bearings and journals heating, and the water service being used only on the

crosshead guides and thrust bearings. After the trial the following parts examined were found to be in excellent condition: after L.P. and H.P. crank pins starboard engine, after L.P. crank pin port engine, and top brass of after L.P. crankshaft bearing on each engine. The crank pin of port I.P. was slightly scored. The contractors had both high-grade lard oil and a selected grade of engine oil on board for lubrication purposes.

Upon the return of the vessel from a trip of about 4,500 miles, the bearings of the main engines were reported in bad condition. On the trip out the engines worked without much noise, but on the return trip the bearings became looser, and the knocking, especially in the high and intermediate on both sides, increased to an excessive degree. The starboard I.P. crank brass and port I.P. slipper were taken down and the babbit found to be in very bad condition, the oilways being almost obliterated. The amount of play in the bearings of the crank pin and guide examined was about $\frac{1}{8}$ inch. There was little trouble with heating in these bearings on this trip. The bearings on the L.P. were not examined at this time, as they had given practically no trouble. A later examination showed the bearings, generally, to be in much worse condition than was at first supposed, and for this reason a more complete examination was ordered. Immediate relining of crosshead slippers, go-ahead eccentrics and crank-pin brasses was found to be necessary. The white metal was found to have dragged in all the brasses, and the oil-ways were obliterated. In view of the fact that the babbit metal withstood the forced-draft trials, whereon at least three times the horsepower of that developed on the recent run was obtained, it was concluded that the primary cause was the use of a poor quality of lubricating oil.

The vessel proceeded to a navy yard for the necessary re-lining of her bearings. Upon arrival an examination of the oil in her tank was made. The amount remaining on hand was about 170 gallons, of which 150 gallons was in one tank and about 20 in another. One sample (b) was drawn from the former

and two from the latter. Of the two latter, one (c) was drawn from near the surface through a pet cock on the gauge glass, and the other (d) through the spigot connected to the pipe leading from the bottom of the tank. A sample (a) of the oil that had been used by the contractors on the preliminary trial on hand was also drawn. Immediately after drawing off, a slightly murky sediment was apparent in all samples, the most being in samples (a) and (b), and the least in the samples (c) and (d), but with very little difference. After standing some sixteen hours, samples (a), (c) and (d) showed a cloudy appearance throughout the bottle, while sample (b), specification oil, looked perfectly clear. There was no apparent increase in the amount of sediment in the bottom of the bottles. A comparison of the contractor's special oil with the specification oil, by rubbing oil in the hand with a little water, showed that the former gives a quicker and more copious lather and works cooler in the hand.

Specification oil was used during the vessel's run, with the exception of 200 gallons of the contractor's special oil used on the starboard engine. The only time that really hot bearings were experienced was once with the valve-stem guide of the starboard I.P. engine, due, probably, to a wrong combination of valves on the assistant cylinders, and once on the port H.P. engine, which was caused by insufficient oil. In both cases the heating was quickly reduced, and no further trouble was experienced. No other bearings developed excessive heat, and the oil would at times show a little lather.

The brasses of the port H.P. engine showed that the white metal at the sides of the brass was squeezed, both in the direction of motion and also in the opposite direction, whereas, if their condition was due entirely to bad oil, it should show only on the go-ahead side. The H.P. and I.P. crosshead slippers showed the most lost motion, although they received a liberal supply of oil.

Of the four samples of oil, the three samples of specification oil showed a decided clearing in the upper part of the bottles and a settling in the bottom, the sample of the contractor's

special oil apparently undergoing the same process, but more slowly.

Examination of the top eccentric strap showed an even wear, with no signs of heating and a squeeze of the white metal in both leading and following edges of the segments.

The cap of No. 6 port main bearing showed the white metal in good condition, but there were two cracks about 6 inches long, extending from the hole for tallow box.

A sample of white metal hammered down and broken showed very nearly the same characteristics as Parsons, and suggested the possibility that the metal was not properly hammered and peened into place.

In connection with the failure of the bearings of *Armored Cruiser D* and *Armored Cruiser C*, comparative compressive tests of three anti-friction metals were made at a navy yard. The test pieces of naval white metal were taken from stock. Parsons white brass showed slightly the best flowing qualities, being pressed thinner than the naval metal, yet showing the least breaking around the edges. Magnolia metal showed a distinct lead color and was about 60 per cent. heavier than either of the others. The metal from which test pieces were cut was first poured. The compressibility of the different metals may be compared by noting the final reduction of heights of approximately equal compression loads.

.....	Naval white brass.		Parsons white brass		Magnolia metal	
Test piece No.....	N–I	N–II	P–I	P–II	M–I	M–II
Original diameter, inches.....	0.749	0.75	†0.749	†0.749	0.743	0.75
height, inches.....	0.75	0.75	0.759	0.758	0.751	0.748
Elastic limit (about), lbs.....	*8,000	8,000	5,000	5,000	5,000	4,500
Final compress loads, lbs.....	36,000	37,000	36,500	36,800	37,000	36,500
Final height, inches.....	0.2	0.192	0.1805	0.18	0.161	0.166

* Less than. † Full.

An opinion was expressed during the work of re-babbitting these bearings that possibly the brasses contained too great a thickness of white metal.

By a comparison of the metal in various bearings, both in *Armored Cruiser C* and *Armored Cruiser D*, it would seem

that those containing about $\frac{7}{16}$ inch of metal (above the ribs) stood their load much better than those containing more than this amount. This might be partly due to the fact that it is more difficult to properly peen the thicker metal in place. None of the crank-pin brasses of *Armored Cruiser C* were fitted with sufficient or proper anchorages for the white metal at the parting of the boxes. The metal in the bearings of this vessel was, for this reason, torn away from the boxes at this point. Similar conditions were found on *Armored Cruiser D*. Dovetailed anchorages were therefore cut in the boxes.

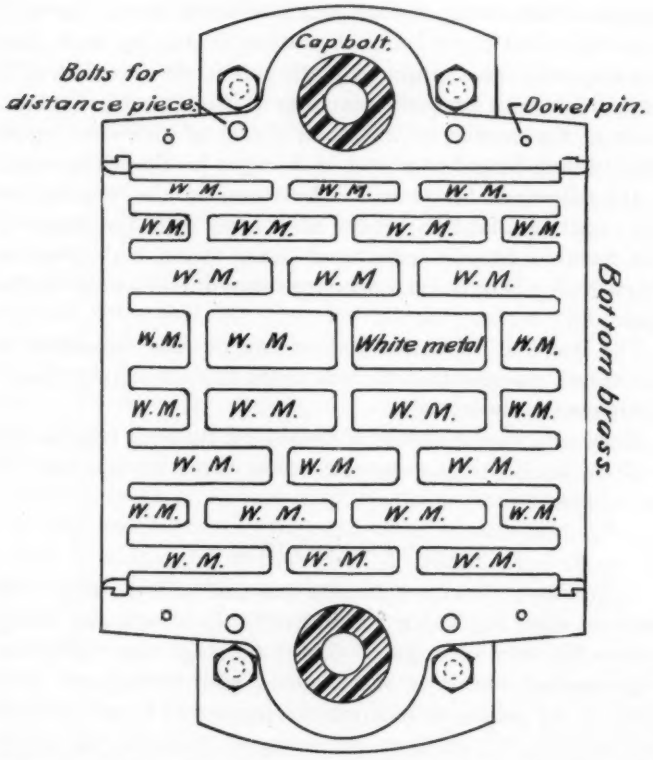
The Board of Inspection and Survey, on final inspection of this vessel, reported that the size of the oil holes in the crank-pin brasses be increased.

Armored Cruiser D was not ready for her final trial, as the work on her bearings remained uncompleted on the date required therefor.

"ARMORED CRUISER E."

Preliminary Trial.—Difficulty was had with the crank-pin bearings only, beginning first in the I.P. bearings, and being worst with the port engine. These bearings were babbitted in the manner shown in sketch No. 1 with white metal composed of 3.7 per cent. best refined copper, 88.8 per cent. of Boustead tin, 7.5 per cent. of antimony, Boustead tin being substituted for Banca tin by special authority of the Bureau.

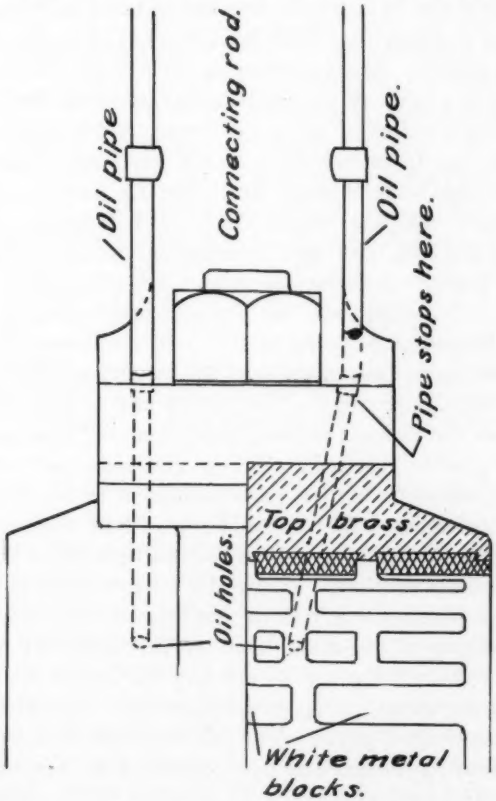
From the sketch it can be seen that the oil channels take up much space, the bearing surface being considerably reduced. On the standardization run, when the engines were speeded up to 130 turns, the I.P. of the port engine developed 5,621 I.H.P., while the H.P. made 2,936, the forward L.P. 2,945, and the after L.P. 2,852. This imposed excessive work on both I.P. engines, and knocking began after the first hour, increasing quickly until most of the metal was pounded out in the port bearing, and the pin was running on the brass, scoring it considerably. All the water possible was kept running on the bearings and there was no sign of overheating,



Sketch No. I.— METHOD OF BABBITTING FOR STANDARDIZATION AND FIRST ATTEMPT AT FOUR-HOUR RUN. METAL POUNDED OUT.

but as soon as the metal began to be pounded down the oil holes in the club end of the connecting rod filled up, and no more oil got into the bearings. The brass plates over the holes through the crank pins were left off, and no grease put in the pin. The oiling gear for the crank-pin bearing consists of four oil tubes led down along the connection rod from the oil cup at top, fed by drip under the cylinder (sketch II). On reaching the club end the oil tube bends out and passes through the club end and projects about $\frac{1}{4}$ inch

down into the hole in crown brass, the hole being carried on through the brass to the bearing (see sketch No. II.) So on each bearing there are four holes in the upper brass, two inboard about 11¼ inches apart, and two outboard 5¾ inches apart. With the type of babbit originally fitted sufficient oil seemed to feed into the bearing, the trouble being caused by excessive horsepower in the I.P. engines and insufficient bearing surface to stand the added strain. The engines are



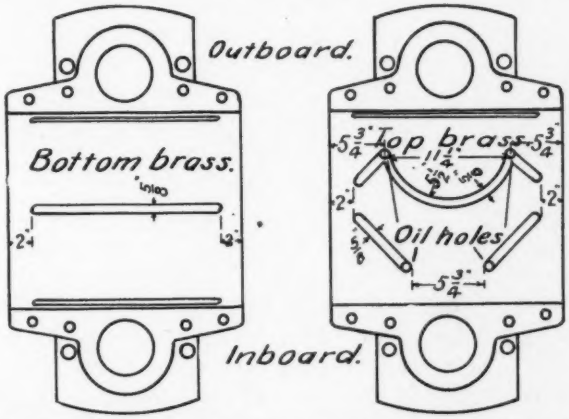
Sketch No. II.—Lead of Oil Pipes to Crank-Pin Brasses. On Inboard Side of Top Brasses it will be Noticed that Oil Holes to Bearing are Farther Apart than on Outboard Side.

designed to have, at moderate speeds, one-third load on H.P., one-third on I.P., and the other third divided between the two L.P. cylinders. When high speeds were reached this did not hold good and the I.P. got most of the work. The crank-pin brasses were fitted to 28 B.W.G. lead.

After this run the spare brasses were fitted on both I.P. pins, and the first attempt was made to run the four-hour trial. In order to lessen the load on the I.P. engine, all L.P. cut-offs were run back to the five-tenths mark on the slot in the arm of the reversing shaft, giving a cut-off in the L.P. of about .55 stroke. After running so for a quarter of an hour they were run back to the eight-tenths mark on the port engine, giving a cut-off of about .35 stroke, and a little later on they were run to seven-tenths on the starboard engine, and these positions were retained until the trial had to be given up. Before the run began the port I.P. crank pin was knocking slightly, and this increased gradually until, after forty-five minutes running, the babbit was all pounded out of the port I.P. bearing, and the pin was again running on the brasses, the pounding being terrific. The starboard I.P. was also knocking, but not so badly. All water possible was kept running on all bearings. Two hours and a half after the trial began the inboard parting piece flew out, wrecking the crank-pit guards, stay rods, platform and water service. Examination showed that the bolts holding it in had broken at the first thread, and that the dowel pins had sheared. The nut on outboard cap bolt had backed one turn and a half, and the set pin was bent up. Practically all the white metal was cut except between the ribs on the brasses. There was no sign of melting of the metal, and the bearings were not hot. In spite of the relief given by altering the cut-off of the L.P. cylinders, the metal had again pounded out. Examination of the brasses showed pounding in the starboard H.P., the starboard forward L.P. and the port forward L.P., but none had heated.

On returning to the ship yard the white metal was melted out of all brasses except the port H.P., and they were re-bab-

bitted with a white metal selected by the builders, supposed to be somewhat harder than the Navy standard composition. In re-babbitting, the bearing was poured practically solid, the oil channels being very much reduced and the bearing surface very much increased (see sketch No. III). This was going to the other extreme and not giving much room for the cir-



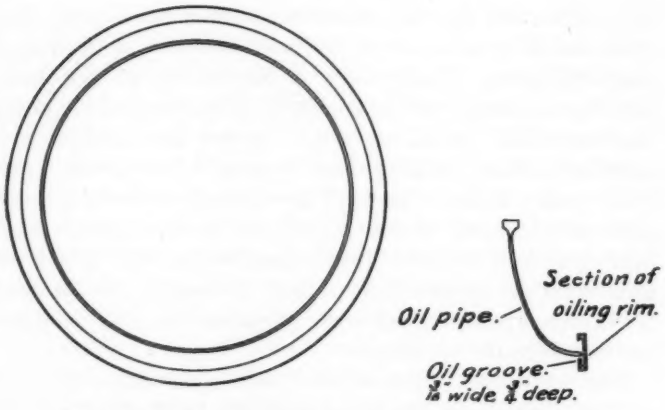
Sketch No. III.— BRASSES AS BABBITTED FOR SECOND, THIRD AND FOURTH ATTEMPTS AT FOUR-HOURS' TRIAL. METAL MELTED OUT UNTIL CENTRIFUGAL OILERS WERE FITTED. OIL CHANNELS ¼″ TO ⅜″ DEEP.

culation of oil in the bearing. The pins were filed down to smooth off the scores, the brasses refitted and the crank-pin holes filled with Albany grease and the brass plates put on. To fill all the pins, 1,120 pounds of this material were necessary. Two one-half inch holes in the pin gave this grease access to the bearing, and centrifugal force forced it out. In addition the regular oiling gear was in use. The L.P. cut-offs were run back to full length of slots in arms of reversing shafts and the brasses were fitted to about 26 lead B.W.G. As soon as the trial began all of the bearings were flooded with water. After running 2½ hours the port forward L.P. crank pin showed evidence of heating, the white metal melting out, and the trial was stopped. The brasses outside were

not so hot that the hand could not be held on them. During this trial the engines worked very smoothly, only a slight knocking being apparent in the port after L.P., and the bearings seemed to be well oiled, from the amount of grease thrown out on the guards. On each revolution the oil seemed to be thrown out and up the oil pipes on the connection rod, against the under side of the cylinder, and with it some of the Albany grease. Examination of the bearings showed the metal in the port forward L.P. melted out to great extent, but the engine was so quickly stopped that there was considerable metal still left. All the Albany grease was out of this pin, and the inside of the pin contained some melted white metal and showed evidence of considerable heat. Very little grease remained in any of the other pins, the greatest amount being 6 pounds. This would indicate that the bearings were all right as regards amount of surface as long as they were lubricated, but so much lubricating material was thrown up out of the oil pipes that very little from that source got into the bearings. With sufficient Albany grease to last the four hours the ship could have made the run, but as 1,120 pounds had been expended in 2½ hours, or at the rate of about 5 tons per day, that system of lubrication was hardly practicable. Still the faults did not seem quite clear, so the vessel was made ready for another attempt. No changes were made in the bearings, though all brasses were gone over, smoothed up, burrs chipped off and the oil holes and channels cleaned out. On the theory that closing up the holes in the crank pins caused the overheating, the brass plates were left off this time and the holes left open, relying only on the regular oiling arrangements. A third attempt was then made at a four-hour trial. On speeding up to the required number of revolutions oil was again thrown out of the oil pipes, and the port forward L.P. crank pin gave evidence of heating. The engines were slowed down and extra water service turned on until the pin was cool. Another attempt was then made, but within about three minutes after the beginning of the trial the port forward L.P. bearing began to get hot, and the trial was called

off. While the ship was returning at slow speed some bath brick and oil were run into the port I.P. bearing to smooth away the scores. Examination of the bearing showed that it had begun to melt, but the engine had been stopped so quickly that very little metal was gone. It was then decided to fit centrifugal oilers, making three systems of lubrication for the crank pins. It would perhaps have been possible to increase, somewhat, the oil channels without sacrificing too much bearing surface, for in all these trials the port H.P. crank pin, fitted with the regular Navy system of bearing blocks made of Navy composition, had never given any trouble whatever, and was in perfect condition.

The centrifugal oilers, as fitted, were composition rims with an open groove in the outer edge, the oil feeding into the groove from an oil pipe led to the oil box on the engine framing. One apparatus was fitted for each crank on the forward side of the web, being secured to it by clamps bolted on the side of the web. When the engine turns over the oil fed into the channel is thrown out toward the rim of the wheel by centrifugal force, and from there led by another oil pipe to a new hole $\frac{3}{4}$ inch diameter bored through the crank pin between the two holes already existing for interior lubrication. As there was very little clearance between the web of the pin as it revolved and the water-service pipes to the main bearings, the oiling rim had to be very thin, and this is its principal fault. At the most it could hold only $\frac{1}{4}$ pint of oil, and might be easily clogged. Then with syringes oil was squirted constantly into the groove during the trials. The brass oiling ring or rim is $3\frac{1}{4}$ inches wide and $\frac{7}{16}$ inch thick. The oil groove in it is $\frac{3}{16}$ inch wide and $\frac{3}{4}$ inch deep. Sketch No. IV shows the details not apparent on sketch VI. After having installed centrifugal oilers the contractors made a run for their own information, keeping the engines at 120 turns for about an hour and a half, everything working well, with the lubrication good and no signs of heating or pounding. The brasses of the intermediate and low-pressure crank pins were removed and overhauled, and found to be in excel-



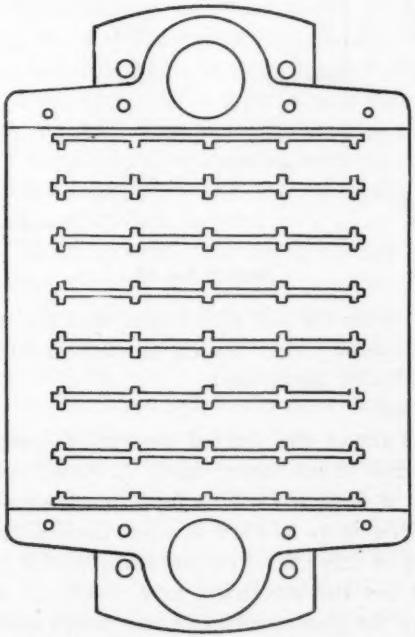
Sketch No. IV.—CENTRIFUGAL OILERS, "CALIFORNIA."

lent condition. In refitting them they were put on very loosely. Leads on the low-pressure crank pins measured from 24 to 27 B.W.G. lead, on the starboard intermediate 22 to 26, and on the port intermediate 21 to 27. In places the leads, which were about 20 B.W.G. originally, showed no contact whatever. The port I.P. crank pin had become so smooth that no traces were left of scoring. The ship was reported ready, and the fourth attempt was made at a four-hour trial. Before the trial began there was a slight knocking in both I.Ps., but this increased only slightly during the trial. Slight knocks developed also in the after L.P. on both sides, but did not increase to any great extent. A great deal of water and oil was run on and into the bearings, and it is probable that the bearings, being continually flooded with both, prevented any increase in the knocking. The ship passed this trial successfully, and the bearings, on examination after the trial, showed little injury. The port I.P. crank pin, which had been scored most on the standardization run and on the first attempt at four-hour run, was worn quite smooth, showing no traces of the former scoring, and was only .003 out of round.

Examination of the pins and brasses showed that greatest stress comes on the bearings during two quadrants of the

revolution, and shows effect in the upper inboard quadrant and the lower outboard quadrant of the bearing.

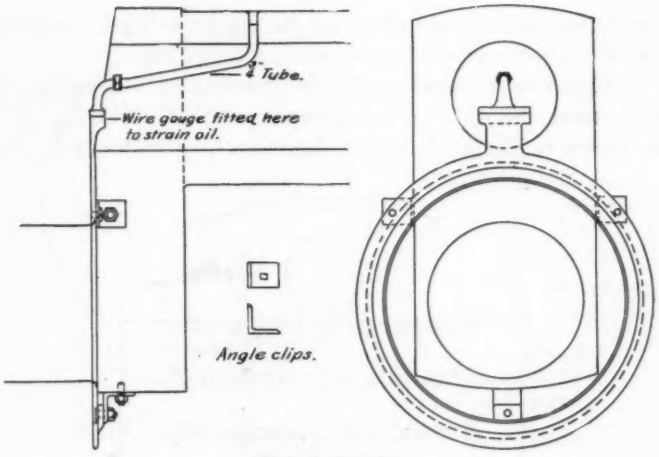
Criticism was passed on the oil grooving of the bearings of this vessel, the opinion being expressed that a bearing such as shown in Sketch V, which has been used with satisfaction



Sketch No. V.—BEARINGS AND OIL CHANNELS OF "NEBRASKA."

in another vessel, would be much more efficient. The opinion was also expressed that the system shown in sketch I would work on all bearings except the I.P., for which a special arrangement would be necessary.

It might be remarked, in conclusion, that the white metal used on the latter trials did not prove superior to the Navy standard metal, as it failed after two attempts of a few minutes each in the third trial for four-hours, but did prove satis-



Sketch No. VI.

factory on the fourth and fifth trials, after the installation of centrifugal oilers. This tended to show faulty lubrication rather than faulty metal used.

To conclude:

On account of the limited amount of space allowed for the installation of machinery in U. S. Naval vessels, the engines must be designed with a minimum amount of bearing surface. This is so to even a more marked degree in our service than in other services, and bears on the question as to whether or not the efficiency, as a whole, of our vessels is increased by the great sacrifice of machinery space for the use of other departments. Under the actual conditions, however, with the high speeds necessary, a high pressure per unit of bearing surface results. Particular care is therefore necessary in designing the bearings and in selecting a suitable material therefor. With perfect design, perfectly even running and perfect lubrication, the kind of material used would to a great extent be immaterial, as, under such conditions the bearing surfaces would be separated constantly by a film of oil, and only fluid friction and the wear of fluid against metal would result. No engine, however, can run absolutely smoothly,

being subject to varying stresses due to thrust, lateral motion and the reversal of the direction of motion, and as a certain amount of grit is ever present, even with the best grades of lubricating oils, the question of the material becomes an important one.

This material must be of such a nature that friction will be reduced to a minimum, and such that it will not heat readily; it must be capable of withstanding the chemical action of the oil, and of oil and salt water; it must be such that it can be easily removed and replaced; it must have a reasonably high melting point, and must stand not only the mean load but the load resultant from intermittent jerks, without deformation; and, finally, it must be such as not to cause undue wear either of the journals or of its own surface.

It seems to be the general practice of marine engineers to use for this purpose one of the alloys known as white metals, meeting, as it does, the necessary requirements of reducing friction to a minimum, of not heating readily, and of being easily removed and replaced. Various metals may be used in the composition of a white metal, the most common of which are tin, copper, antimony, lead and zinc. The alloy selected by the Bureau of Steam Engineering for the main bearings of naval vessels is, as stated at the beginning of this article, made up of copper, tin and antimony. The copper is introduced to give a high melting point, the tin for strength, and the antimony for hardness. The tin and antimony together give the metal stiffness, while the tin and copper tend to minimize chemical action. To give the necessary strength it will be noted that a high percentage of tin is used; the antimony, being more or less brittle, a limited amount is used to supplement the strength by hardness; only enough copper is used to give the desired melting point. Lead is the best wearing metal known, and being very soft and plastic, forms an excellent bearing surface, but, because of the difficulty in overcoming its tendency to flow, it is omitted from the above-mentioned alloy. Zinc adheres to iron when even slightly heated, and becomes brittle under the effect of heat, so is

little used. The standard Navy metal is generally accepted as one of the very hardest alloys that can be classified as a white metal.

The compressive test, such as was made of the metal from the bearings of *Armored Cruiser D*, is of great importance, as it indicates whether the alloy is hard enough to support the required load without deformation, whether it is so brittle that it will split under its load, and whether sufficiently plastic to run cool under the varying stresses to which it is subjected in actual service.

Great care is necessary in the mixing and casting of the metal originally, as well as in the re-melting, re-pouring and peening when the bearings are made. The most expert hands are required for this work as a special practical knowledge is necessary. It is of vital importance that the metal be carefully peened in place in order to compress and harden the metal and fill up all crevices, after which it must be accurately bored and scraped to a proper fit on the journal.

There is probably little difference in the value of the generally accepted metals by reason of their composition, as the good or bad bearing undoubtedly depends more upon the re-melting, re-pouring and peening than on the metal itself, always assuming that the bearing is properly adjusted and the lubricant good.

In the design of the bearing there must be considered—the construction of the bearing, whether the white metal should be in one slab or more, and of what thickness, the clearance to be allowed, the arrangement and extent of oil grooves, the means of supplying the oil, and the point of admittance of the oil; of late, the question of the necessity of centrifugal oilers has come up.

From the above conclusion it can be seen that there are a multitude of points bearing upon the efficient performance of main engine bearings in service. Faults of one nature may give rise to difficulties at high speeds, whereas others may even cause failures at the lower speeds. It is a matter of great difficulty, in cases such as are cited in this article, to

pin down the cause to but one of the many reasons, for, to sum up once again some of the causes for hot bearings:

(1) Insufficient area of bearing surface;

(2) Strength of the casing;

(3) Wearing surfaces out of line or badly fitted;

(4) Improper adjustment;

(5) Improper or insufficient lubrication, due to poor means of supply, poor oil, presence of dirt or grit, or to neglect;

(6) Unsuitable bearing metal.

It will be seen that the design, the adjustment, and the management are all three jointly concerned.

It has not been the object of the writer to put forth herein his reasons for the causes of the failures cited in the cases of the five vessels mentioned, but to present the facts of each case before the readers, that they may draw their own conclusions.

It seems only just to close these remarks with the statement that our engines are required in many instances to develop a much greater horsepower than the specifications of the vessel have called for, in order to attain the speed required by the contract. The uncertainty due to such conditions, coupled with the fact that so limited an amount of space is allowed for the installation of the machinery, requires unusually careful judgment in the design of our bearings.

Some time has elapsed since the writer began this article, and in the meantime the various defects noted have been remedied by re-adjustments and the use of high grades of lubricating oil. In consequence thereof there have been no adverse reports on the working of the bearings.

U. S. REVENUE CUTTER *ITASCA*.

THE REHABILITATION OF THE U. S. S. *BANCROFT*.

By First Assistant Engineer, Charles S. Root, U. S. R. C. S., Member.

A steam vessel having become desirable for use in training cadets of the line and engineer corps of the Revenue Cutter Service, to replace the bark *Chase*, the U. S. S. *Bancroft* was provisionally transferred from the Navy to the former service on January 20, 1906. On June 30th of the same year the transfer was confirmed by act of Congress. After the final transfer the vessel was rechristened *Itasca*.

For the information of those unfamiliar with the vessel the following description is given.

Intended for a practice ship for the Naval Academy, the *Bancroft* was designed to contain everything in the shape of machinery and equipment that was usually found on a large protected cruiser, and this scheme was followed out even to the extent of fitting steam turning gear to her diminutive propelling engines.

She was built by the Samuel L. Moore and Sons Co., of Elizabeth, N. J., and cost when fully equipped and ready for sea, $431,281.63. Her rig was that of a barkentine to topgallant sails without a bowsprit. She was built with poop and topgallant forecastle, ram bow, protected rudder and steering gear.

A light protective deck extended from stem to stern, with a cofferdam for cellulose from forward of the forward magazine to aft of the after magazine, including the machinery and boiler spaces. No double bottom was provided, but all other watertight sub-divisions were fitted that were usually found in vessels of the *Baltimore* class.

The armament consisted of four 4-inch R.F. rifles, two 6-pounder Hotchkiss, two 3-pounder Driggs-Schroeder, one 1-pounder Hotchkiss, one 37-mm. Hotchkiss, one Gatling gun and two torpedo tubes discharging above water.

She was launched April 30, 1892, and the official four-hour trial was held January 26, 1893.

MACHINERY.

The boilers, two in number, were of the gunboat type. They were placed side by side in one watertight compartment and were fired from the after end. The uptakes discharged into one funnel. Forced draft was installed on the Kafer closed ash-pit system. The main feed pump was in this compartment.

The engines, two in number, were the usual three-cylinder, inverted, direct-acting type, with the H.P. cylinders forward and the L.P. cylinders aft. The H.P. and M.P. cylinders were fitted with piston valves and the L.P. with an ordinary double-ported slide valve. All valves were worked by Stephenson link motion with double-bar links. The bedplates were cast in three sections. The bedplates, pistons, crossheads, cylinder- and valve-chest covers were of cast steel. The cylinders were supported on wrought-steel columns with steel tie rods.

Steam-hydraulic reversing gear was provided. The crank shaft was in three interchangeable sections. The shafting was all of mild steel and hollow.

The propellers, of manganese-bronze, were built up, three-bladed true screws and turned outboard.

The main condenser was cylindrical with a composition shell, set abaft the propelling engines and in the center of the vessel, the tubes being athwartship. A small auxiliary condenser with air and circulating pumps was installed, for use when not under way.

An air pump of the twin-cylinder type was fitted, and also a centrifugal circulating pump for the main condenser, together with the usual naval outfit of auxiliaries.

In the original plans of this vessel there were nine bunkers below the protective deck, one on each side of the forward magazine, one cross bunker forward of the boilers, two bunkers on each side of them and one on each side of the engine room. In addition, there were eight bunkers between the protective and berth decks so arranged as to trim into the lower bunkers. The capacity of the bunkers proper was about 115 tons.

After the transfer of the vessel to the Revenue Cutter Service, on February 8, 1906, a board of survey was appointed, with orders to recommend such repairs as would be necessary to put the old steam machinery of the vessel in efficient condition for sea service. The report of the board showed that most of the machinery—with the exception of the propelling engines—was practically worn out and not worth repairing. Upon this report the Department decided that it would be necessary to install practically new machinery throughout, with boilers of a different type. Extensive repairs to the hull were also planned. The contract for rebuilding was awarded to the Newport News Ship Building and Dry Dock Company, the total cost of which amounted to about $120,000. The result is that the vessel is now practically as good as new, and in many respects better than when she was originally designed. To reproduce her, not including her battery and equipment, would cost the Government at least $350,000, at the present prices for labor and material.

REBUILDING.

The old battery and the sponsons for the 4-inch guns were removed and the following guns are now mounted: Four 6-pounder Driggs-Schroeder, two 3-pounder Driggs-Schroeder, six 3-pounder Hotchkiss, two 1-pounder Hotchkiss, and two 1-pounder field guns.

The rig, which in September, 1896, had been changed from that of a barkentine to two pole mast, was replaced by that of an hermaphrodite brig. The dimensions follow:

Foremast :	
Lower mast, heel to hounds, feet and inches	44–0
Head, feet and inches	7–6
Fore topmast, feet and inches	38–9
Rake, degrees	5
Mainmast :	
Heel to hounds, feet and inches	46–0
Head, feet and inches	7–6
Main topmast, feet and inches	38–0
Rake, degrees	5½
Fore topgallant yard, length over all, feet and inches	27–0
topsail yard, length over all, feet and inches	40–0
yard, length over all, feet and inches	53–0

The standing rigging is all of galvanized iron.

All old wooden decks were removed and replaced with new. A new pilot house and chart room were fitted.

The protective deck over the engine and boilers was removed and trunks carried up to the spar deck, large openings being left for light and ventilation.

The bunker bulkheads in the engine room were removed and this space utilized for auxiliary machinery. A cross bunker was built between the engine and boiler space to make up the coal capacity lost by removal of the engine-room bunkers. At each side and directly outboard of this cross bunker fresh-water tanks are built in, with a total capacity of 4,200 gallons. The capacity of the lower bunkers is now about 117 tons, a little in excess of what it formerly was.

This arrangement has increased the floor space around the machinery. The fireroom is light and airy, and is 10 feet long by 19 feet wide. The new arrangement of the vessel is appended. Plates II and III.

BOILERS.

The two original boilers, of the horizontal, fire-tube gunboat type, had two corrugated furnaces each, with a separate combustion chamber for each furnace. The following are the principal data :

Number of boilers	2
Steam pressure, pounds	160
Length, feet and inches	17–0
Diameter, outside, feet and inches	8–9½
Grate surface, total for two boilers, square feet	87.75
Heating surface, total for two boilers, square feet	2,686.0
Heating surface ÷ grate surface	30.61
Furnaces, mean internal diameter, inches	41
Number of tubes, one boiler	314
Length of tubes between tube sheets, feet and inches	7–0
Outside diameter of tubes, inches	2
Area through tubes, one boiler, square feet	5.44
Grate surface ÷ tube area, square feet	8.06
Diameter of funnel, feet and inches	4–7
Height of funnel above grate, feet	40

These boilers were removed, together with their uptakes, funnel and all other fittings. They were replaced by Babcock & Wilcox boilers of the following dimensions:

Number of boilers	2
Drums, length of shells, feet and inches	7–4⅛
inside diameter, feet and inches	3–0
Drum plates, thickness, inch	0½
Heating surface, two boilers, square feet	3,825
Grate surface, two boilers, square feet	85
H. S. ÷ G. S.	45
Funnel, height above grates, feet and inches	48–0
inside diameter, feet and inches	5–0
Generating tubes, outside diameter, inches	2 and 4
thickness of 2-inch	No. 8 B.W.G.
4-inch	No. 6 B.W.G.
exposed length, feet	9
inclination of, degrees	15
number of 2-inch, one boiler	314
4-inch, one boiler	12
Height of boiler, base to drum center, feet and inches	9–5⅛
casing top, feet and inches	10–1½
Width of boiler, feet and inches	7–3½
Length of boiler at base, feet and inches	10–3½

The pressure parts of these boilers are of mild steel and the tubes cold drawn. All details are up to the usual Government standard. The boilers are placed abreast, and well forward in the old boiler compartment. The gases discharge

into a single funnel. There is a space, 30 inches wide and of ample height, between the boilers, for passing coal from the forward bunkers.

FORCED DRAFT.

The original boilers were fitted with forced draft on the closed ash-pit system. Air was supplied by two fans bolted to the coal-bunker bulkheads and taking air from ducts over the tops of the boilers.

Diameter of cylinders (two), inches	3½
Stroke, inches	2½
Diameter of fan, inches	36
Width of fan, inches	12
Diameter of inlet, inches	22

These fans were removed and one fan of the Sturtevant make and of the following dimensions installed:

Diameter of cylinders (two), inches	5
Stroke, inches	4
Diameter of fan, inches	60
Width of fan at periphery, inches	17
Diameter of inlet, inches	42

The fan engine is located in the upper engine room, the fan being outside the engine-room casing on the berth deck. It takes air from the crew space direct and discharges through a 20-inch by 48-inch duct, below the berth deck, to the fireroom. Air valves and ducts are also connected to the fan casing and living quarters for exhausting vitiated air.

PROPELLING ENGINES.

The cylinders were 13½, 21 and 31 inches diameter and 20 inches stroke. In order to increase the economy and steaming radius of the vessel, it was decided to raise the steam pressure from 160 to 215 pounds and cut down the high and intermediate cylinder diameters.

New H.P. cylinders, 12 inches diameter, and new M.P. cylinders, 19 inches diameter, complete with new pistons and

valves, were made for each engine, and a very light cut was taken through each L.P. cylinder. The piston rods and valve stems were either renewed or resurfaced and new packing for pistons, piston rods and valves stems was furnished.

New crossheads, crosshead pins and crosshead bearing brasses were made, and the bar guides were taken down and planed on all four faces.

The crank and thrust shafts were removed and trued up on all bearing surfaces, and brasses and bearings were refilled with white metal, refitted and lined up. The stern-tube and bracket bushings were restaved with lignum vitae. All eccentric sheaves were trued up and the straps remetaled and bored out. The reversing cylinders and gear were overhauled and the oil cushioning cylinders and hand pumps removed.

New balanced throttle valves were fitted in place of the old gridiron valves. New oil and water services, gauges and revolution counters were also fitted.

The following are the principal data for the main engines as now installed:

Number of engines	2
Cylinders, number for each engine	3
diameter of H.P., inches	12
M.P., inches	19
L.P., inches	31
Stroke of pistons, inches	20
Ratio of the net piston areas, at present	1 : 2.5 : 6.8
old engines	1 : 2.4 : 5.3
Valves, diameter of the H.P., inches	6
M.P. (two), inches	6
length of the L.P., feet and inches	2–0⅞
width of the L.P., feet and inches	2–3
Piston rods, diameter, inches	2½
Main steam pipe, diameter, inches	4½
Exhaust to M.P., diameter, inches	5½
L.P., diameter, inches	7½
condenser, diameter, inches	9½
Clearance in H.P. cylinder (volumetric), per cent	11.67
M.P. cylinder (volumetric), per cent	12.0
L.P. cylinder (volumetric), per cent	7.1
Connecting rods, length, center to center, inches	40
diameter at upper end, inches	2½
lower end, inches	4

Crosshead, surface of slipper, square inches	71.8
pins, diameter, inches	3¼
Crank shaft, diameter, inches	6
axial hole, inches	3
journals, diameter, inches	6
length, inches	5¼
length of each section, feet and inches	2-7½
Crank pins, diameter, inches	6
length, inches	6
axial hole, inches	3
Thrust shaft, diameter, inches	5¼
axial hole, inches	2½
collars, diameter, inches	9
number for each shaft	7
surface, total for one engine, square inches	263.55
Line shaft, diameter, inches	6
axial hole, inches	3
Propeller shaft, diameter, inches	6
axial hole, inches	3

PROPELLERS.

The propellers are of manganese-bronze with three elliptical blades, and turn outboard. The blades can be set for pitches between 6 feet 9 inches and 8 feet 3 inches. The boss is secured to the shaft by a taper key and a composition nut. The end of the shaft is covered by the usual cap.

PROPELLER DATA FOR THE TRIAL.

Number propellers	2
Diameter over tips of blades, feet and inches	7–0
of hub, feet and inches	1–10
Pitch as set for trial, feet and inches	7–9
Helicoidal area, square feet	15.0
Projected area, square feet	10.66
Disc area, square feet	35.85
Pitch ÷ diameter	1.1
Helicoidal area ÷ disc area	0.418
Projected area ÷ disc area	0.297
Immersion of shaft centers at mean draught on trial, feet and inches	7–8½

CONDENSERS.

The main-condenser shell is cylindrical, of composition and in one piece. The circulating water enters at the bottom and leaves at the top, the water flowing through the tubes.

The tubes and tube sheets were removed and the shell thoroughly cleaned. The old tube sheets were replaced and new tubes and ferrules fitted.

Diameter of the shell, feet and inches	4–0
Length over heads, feet and inches	7–8¼
Tubes, outside diameter, inch	0⅝
length between tube sheets, feet and inches	6–3½
number of	1,772
Cooling surface, square feet	1,824

There was, besides, an auxiliary condenser, with an air and circulating pump, all of the Davidson type. As valves were fitted in the exhaust pipes from the main engines, this condenser was deemed to be unnecessary and was removed from the vessel.

AIR PUMP.

The air pump, of the Davidson twin-cylinder type, was thoroughly overhauled.

Diameter of the steam cylinder, inches	8
water cylinders (two), inches	14
Stroke, common, inches	12

CIRCULATING PUMP.

The circulating pump, of the usual centrifugal type, driven by a vertical steam engine, was overhauled and put in good order.

Diameter of steam cylinder (one), inches	6
Stroke, inches	5
Diameter of impeller, inches	22
Width of impeller at hub, inches	4½
tips of blades, inches	2
Diameter of inlet nozzle, inches	8
outlet nozzle, inches	7

FEED AND FILTER TANK.

The old feed and filter tank was removed from the vessel and a new tank, placed in a more accessible position, is now located on the port side.

PUMPING OUTFIT.

The following pumping outfit was removed from the vessel together with the old pipe system.

.....	Kind.	Cylinders.		Stroke.
		Steam.	Water.	
		inches.	*inches.*	*inches.*
Main feed	vertical single	9	5¼	10
Auxiliary feed	vertical single	9	5¼	10
Water service	vertical single	9	5¼	10
Fire and bilge	vertical single	12¼	7¾	12
Distiller circulating..	vertical single	5¼	5	8
Evaporator feed and tank	vertical single	3	1¾ / 1¼ tank.	4

The new pumps fitted are as follows:

.....	Kind.	Cylinders.		Stroke.
		Steam.	Water.	
		inches.	*inches.*	*inches.*
Main feed	horiz'l duplex	7¼	4¼	10
Auxiliary feed	horiz'l duplex	8	5	12
Water service	vertical single	7	7	10
Fire and bilge	vertical single	12	7	12
Distiller circulating..	vertical single	4¼	6	6
Evaporator feed	vertical single	3¼	2¼	4

The feed tank and horizontal pumps are placed well out in the wings in the space formerly occupied by the engine-room coal bunkers.

FEED-WATER HEATER.

A new feed-water heater with spiral tubes and about 90 square feet of heating surface was installed, the heating agent being the exhaust steam from the auxiliaries. No heater was fitted in the old feed system.

DISTILLING APPARATUS.

The old Baird evaporator, No. 3, Type C, being worn out and beyond repair, was removed from the vessel. A new evaporator and distiller of the Quiggins type, with spiral tubes, is now installed. The maximum capacity is 2,000 gallons per 24 hours.

OFFICIAL TRIALS UNDER FORCED DRAFT.

Draught, molded, at beginning, forward, feet and inches.....
aft, feet and inches.....
mean, feet and inches.....
end, forward, feet and inches.....
aft, feet and inches.....
mean, feet and inches.....
mean, for trial, feet and inches.....
Displacement at mean draught as above, tons.....
Average speed in knots per hour.....
Slip, mean of both screws, in per cent. of their own speed.....
Revolutions of the main engines, per minute.....
mean of both engines.....
Piston speed, feet per minute.....
Steam pressure at boilers (gauge).....
M.P. receiver (absolute).....
L.P. receiver (absolute).....
Vacuum in condenser (inches of mercury).....
Opening of throttle.....
Cut off, decimals of stroke from beginning (mean of both ends), H.P.....
M.P.....
L.P.....
Double strokes of air pump, per minute.....
feed pump, per minute.....
Revolutions of circulating pump, per minute.....
Temperature, engine room, degrees Fahrenheit.....
injection, degrees Fahrenheit.....
discharge, degrees Fahrenheit.....
hot well, degrees Fahrenheit.....
feed, degrees Fahrenheit.....
fireroom, degrees Fahrenheit.....
Revolutions of blowers, per minute.....
Mean effective pressure, H.P. cylinder, pounds.....
M.P. cylinder, pounds.....
L.P. cylinder, pounds.....
referred to L.P. cylinder, pounds.....
Indicated horsepower: H.P.....
M.P.....
L.P.....
Collective, each main engine.....
both main engines.....
Air pump.....
Circulating pump.....
Feed pump.....
Blowers (estimated).....
Other auxiliaries (estimated).....
Collective, all auxiliaries.....
all machinery in operation.....
Indicated thrust, main engines only, pounds.....
per sq. ft. developed area of propeller (15 sq. ft.), pounds.....
inch of thrust bearing (263.55 sq. inches), pounds.....
Square feet of cooling surface in condenser per I.H.P.....
heating surface in boilers per I.H.P.....
I.H.P. per square foot of grate surface.....
Kind of coal.....

U. S. S. "BANCROFT." Official trial, Jan. 26, 1893.			U. S. S. "ITASCA." Official trial, June 14, 1907.		
Starboard.		*Port.*	*Starboard.*		*Port.*
.....	11–00	.....	.....	9–05	.....
.....	12–00	.....	.....	12–06	.....
.....	11–06	.....	.....	10–11¼	.....
.....	10–10	.....	.....	9–04	.....
.....	11–11	.....	.....	12–05	.....
.....	11–04¼	.....	.....	10–10¼	.....
.....	11–05¼	.....	.....	10–11	.....
.....	832.0	.....	.....	775.0	.....
.....	**14.87**	.....	.....	**15.28**	.....
.....	15.5	.....	.....	17.14	.....
223 09	.....	221.77	242.25	.....	240.0
.....	222.39	.....	.....	241.125	.....
743.5	.....	739.2	808.5	.....	800.0
.....	167.6	.....	.....	206.13	.....
64.32	.....	65.76	99.7	.....	96.7
24.1	.....	25.76	25.7	.....	26.7
.....	26.34	.....	.....	23.33	.....
Wide.	.....	Wide.	Wide.	.....	Wide.
.72	.....	.72	.68	.....	.68
.72	.....	.72	.70	.....	.70
.72	.....	.72	.70	.....	.70
.....	75.5	.....	.....	50.0	.....
.....	35.25	.....	.....	31.0	.....
.....	236.31	.....	.....	354.5	.....
.....	76.2	.....	.....	90.0	.....
.....	44.0	.....	.....	66.0	.....
.....	87.7	.....	.....	.....	.....
.....	90.6	.....	.....	112.0	.....
.....	119.9	.....	.....	226.0	.....
.....	77.5	.....	.....	93.5	.....
574.0	.....	580.0	.....	513.0	.....
61.98	.....	65.87	91.0	.....	103.25
21.77	.....	22.93	42.9	.....	39.3
13.98	.....	12.48	13.3	.....	14.25
34.61	.....	35.28	42.7	.....	44.1
197.0	.....	208.04	246.6	.....	276.8
169.09	.....	176.52	294.9	.....	267.7
222.33	.....	210.36	244.7	.....	259.7
588.42	.....	594.92	786.2	.....	804.2
.....	1,183.34	.....	.....	1,590.4	.....
.....	5.09	.....	.....	3.4	.....
.....	1.35	.....	.....	3.0	.....
.....	1.7	.....	.....	2.2	.....
.....	12.0	.....	.....	15.0	.....
.....	9.5	.....	.....	10.0	.....
.....	29.64	.....	.....	33.6	.....
.....	**1,212.98**	.....	.....	**1,624.0**	.....
11,232.0	.....	11,422.0	13,819.0	.....	14,268.0
748.7	.....	761.4	921.26	.....	951.2
42.5	.....	43.22	52.43	.....	54.14
.....	1.54	.....	.....	1.123	.....
.....	2.18	.....	.....	2.355	.....
.....	13.82	.....	.....	19.2	.....
Selected Pocahontas.			Georges Creek, mostly slack.		

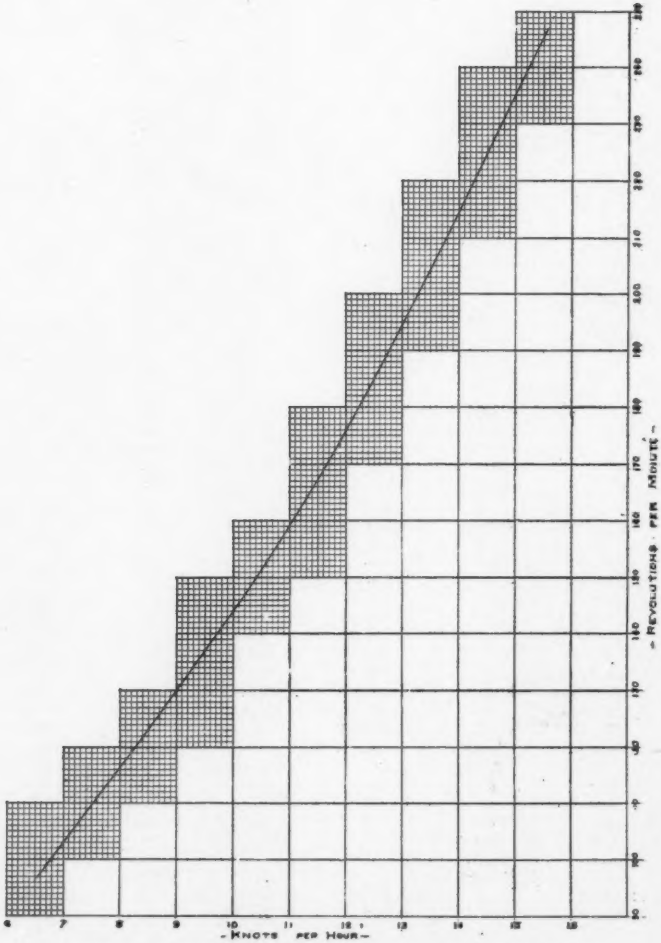
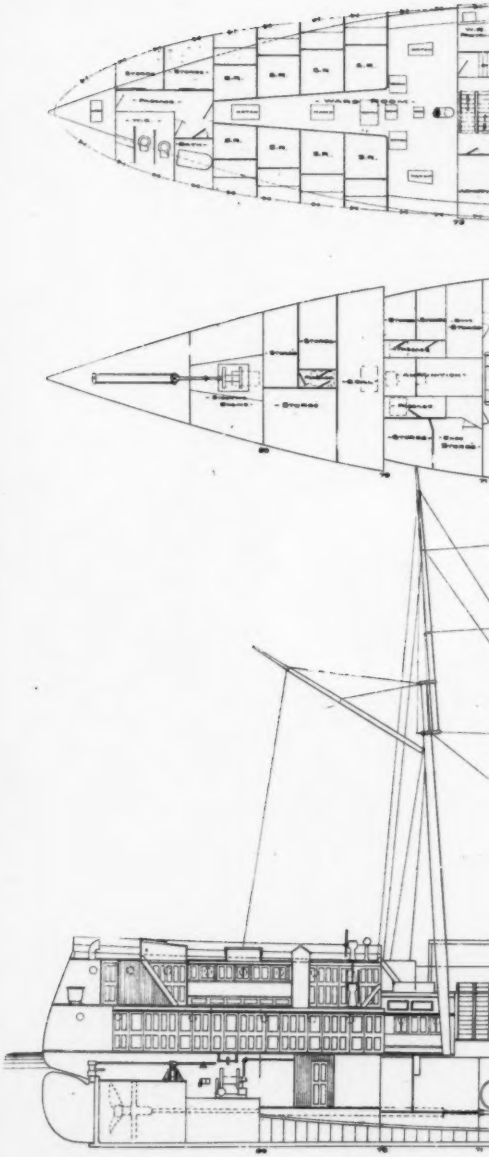


Plate I.—STANDARDIZATION CURVE, U. S. S. "ITASCA."



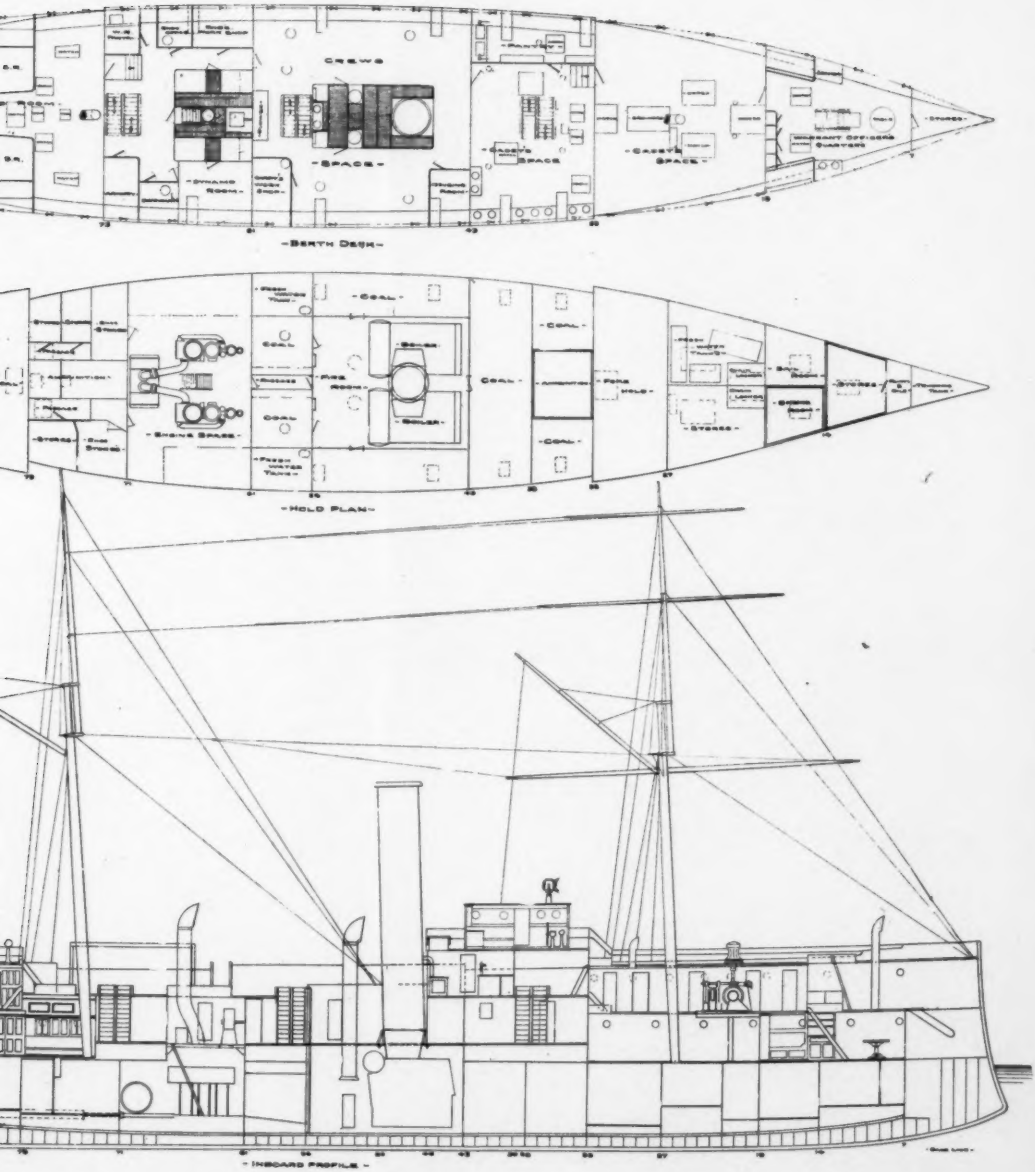
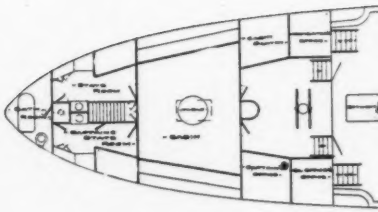
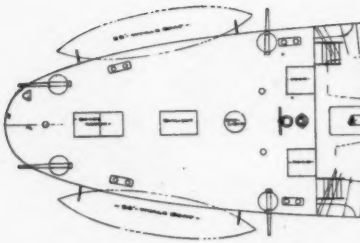
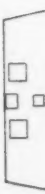


Plate II.—U. S. S. "ITASCA."







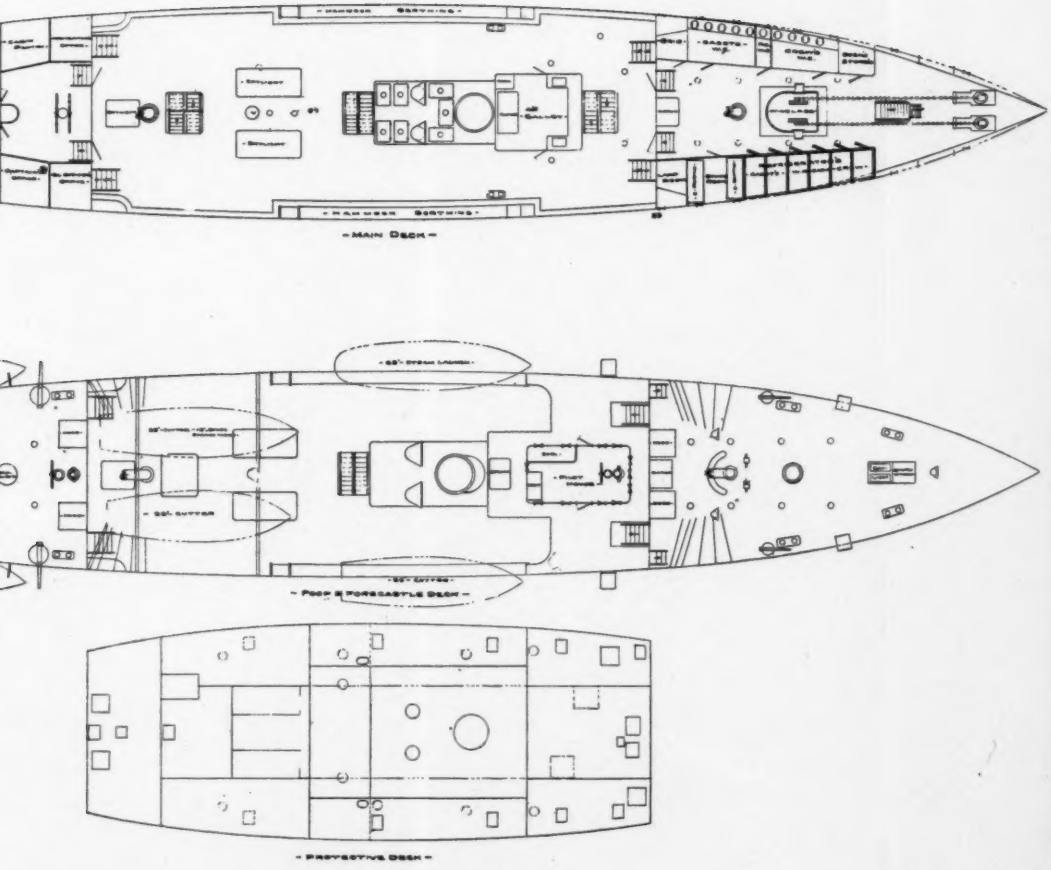


Plate III.—U. S. S. "ITASCA."

VENTILATING APPARATUS.

In the engine room there was formerly a small steam-driven fan for ventilating the living quarters. This fan was removed and the air ducts connected to the forced-draft fan on the berth deck.

OTHER AUXILIARIES.

The steering engine is placed well aft, near the tiller and beneath the protective deck. It is of the Williamson type, with steam cylinders 6 inches by 6 inches. This engine was overhauled. The windlass engine, of the Providence type, was fitted with new brasses and operating valves.

ELECTRIC PLANT.

The electric plant consisted of two 4-kw. Thompson-Houston dynamos delivering current at a pressure of 80 volts. They were driven by Armington & Sims' vertical engines at 500 revolutions per minute. Each engine had two cylinders, 5 inches diameter and 3 inches stroke. These units being worn out, were removed.

The new generating set is of the General Electric make. The generator has six poles and delivers 15 kw. at a pressure of 125 volts at the switchboard. The engine is driven by a single steam cylinder, 8 inches diameter and 6 inches stroke. The set averaged 408 revolutions per minute on trial.

The old wiring and fixtures were removed and entirely new systems installed. All wires are in metal conduits and most of the fixtures are steamtight. The vessel is fitted with 202 sixteen-candlepower lamps, a telephotos, truck lights and a General Electric 18-inch searchlight. New general alarms and call bells were fitted throughout.

THE TRIALS.

On June 13, 1907, the dock trial was made and, everything working successfully, the standardization and four-hour trials were run on the following day.

The vessel left the ship yard at 10·40 A. M. and made several runs over the measured mile off Newport News. The

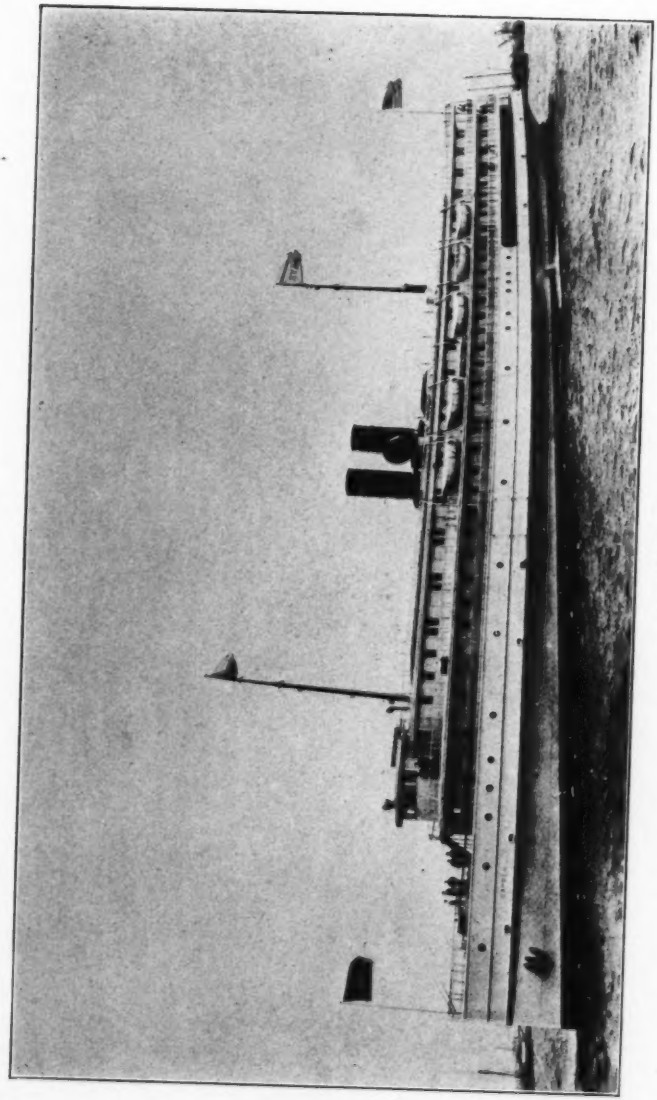
results checked exactly with the original standardization curves.

The endurance trial, under forced draft, commenced at 1·00 P. M. and ended at 5·00 P. M. There was a moderate beam sea and a fresh breeze. The course was from the Thimble Light off Old Point Comfort to the sea buoy off Cape Henry. The vessel was trimmed too much by the stern, and though the displacement was slightly less than on the original trial, the indicated horsepower for any given speed of the propelling engines was from five to ten per cent. higher than on the original trial. The trial was successful in every way. The boilers steamed freely in spite of the fact that the coal was of very poor quality. The water level in the boilers was steady at all times.

The main engines and auxiliaries performed in a satisfactory manner. No water was used on any of the bearings. The dynamo was in operation during the entire run.

The turns were made at full speed at each end of the course, both throttles being wide open for the entire four hours. Live steam was frequently admitted to the M.P. and L.P. receivers. On this account it is impossible to make a comparison, for steam consumption, between the vessel's performance with the old and new cylinder ratios.

The *Itasca* is now on a European cruise, and it is expected that sufficient data will have been obtained, by the time she returns, to make an interesting comparison. A synopsis of the vessel's official trials of January 26, 1893, and June 14, 1907, is appended, Table and Plate I.



S. S. "CAMDEN."

THE TURBINE STEAMSHIP *CAMDEN*.

DESCRIPTION AND TRIALS.

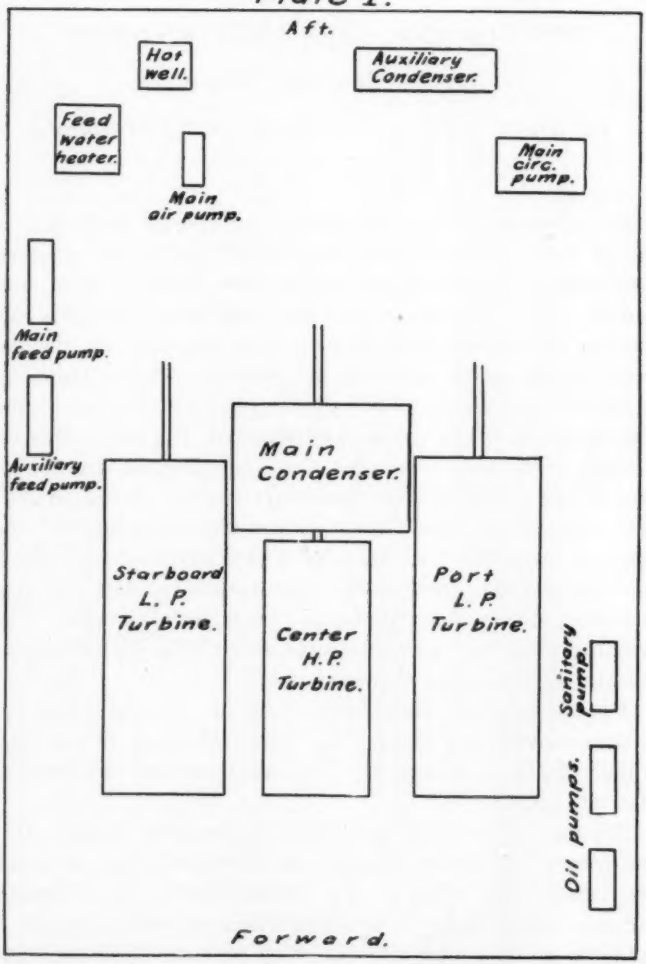
By Lieut. A. F. H. Yates, U. S. N., Member.

The *Camden*, built by the Bath Iron Works, Bath, Me., is one of two passenger ships contracted for by the Eastern Steamship Company for service between Boston, Mass., and Bangor, Me. The contract for the construction of the *Camden* was concluded June 1, 1906, and provided for the delivery of the vessel on or before June 15, 1907, a bonus of $300 per day being allowed for delivery in advance of the date specified. The guaranteed speed of the vessel was an average of at least seventeen knots an hour on six double runs of the vessel over her passenger route. Owing to market conditions in the United States the fulfillment of the contract within the time specified would have been extremely doubtful had not contracts for the machinery, steel castings and steel drums been promptly placed in England. The principal steel castings and drums were delivered within five months from the date of the contract.

The keel of the *Camden* was laid on July 19, 1906, the hull launched on February 14, 1907, the trials of the machinery finished on May 23, 1907, and the vessel delivered on May 31, 1907.

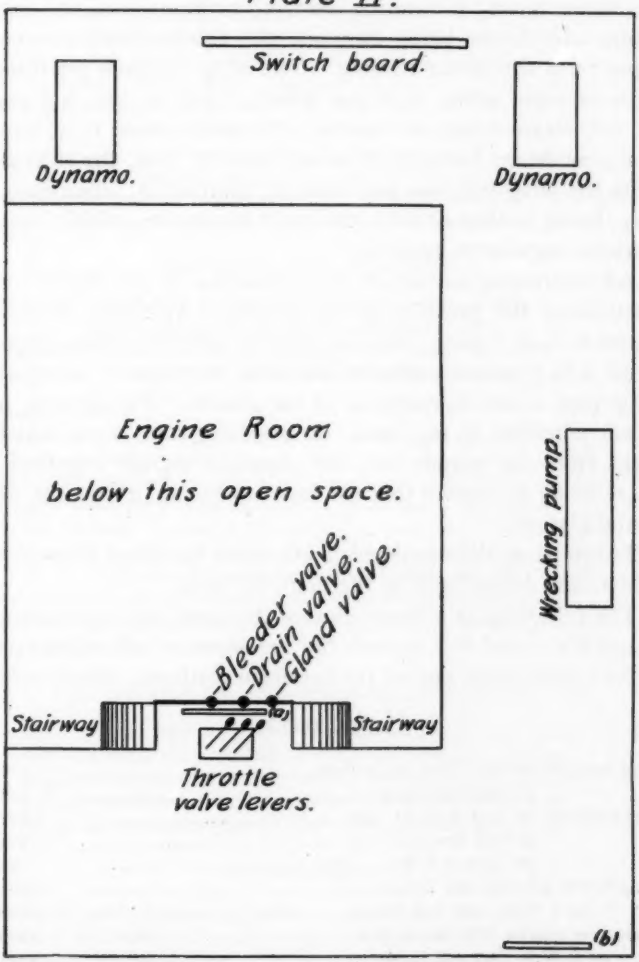
The machinery trials were held at Boothbay Harbor, Me., and were of particular interest, the *Camden* being one of the first vessels to be fitted in the United States with a Parsons turbine installation. The installation on the *Camden* is made up of one center and two wing turbines of about 4,000 brake horsepower, making upwards of five hundred revolutions per minute. The center turbine is the high-pressure ahead turbine, and takes its steam direct from the boilers. The wing turbines are low-pressure and are run ahead on the

Plate I.



SHOWING ARRANGEMENT OF MACHINERY IN ENGINE ROOM, STEAMSHIP "CAMDEN."

Plate II.



(*a*) Frame for the following gauges: main steam pressure; starboard astern pressure; port astern pressure; starboard ahead pressure; port ahead pressure; starboard revolution indicator; center revolution indicator; port revolution indicator; vacuum, main condenser; clock.

(*b*) Frame for the following gauges: auxiliary steam pressure; auxiliary exhaust pressure; steam heat; oil pressure; gland pressure.

SHOWING ARRANGEMENT OF ENGINE-ROOM PLATFORM OVERHEAD, STEAMSHIP "CAMDEN."

exhaust steam from the center (high-pressure) turbine. No live steam is used in running the wing turbine ahead, the only steam admittance being through the thirteen-inch receiver pipes from the center turbine. The wing turbines are fitted with separate astern turbines, which, when in use, are run on live steam from the boilers. Although there is an auxiliary condenser installed it is only used in part, the exhaust from the wing turbines and from all auxiliaries, when under way, being connected with the main condenser, which has a Parsons augmentor attached.

An interesting feature of the installation is the method of controlling the pressure in the glands. Auxiliary exhaust steam is used entirely, and the pressure on the various glands (from 2 to 3 pounds) remains the same, by reason of an equalizing pipe which connects all of the glands. The amount of steam admitted is regulated by watching the vapor which leaks from the glands into the atmosphere, the regulating valve being so located that the man operating it can see all of the glands.

Lubricating oil is pumped to the main bearings through a cooler, and drains back to the main oil tank.

The following is a description of the vessel and machinery. Plates No. 1 and No. 2 show the arrangement of machinery in the engine room and on the overhead platform, respectively.

HULL DATA.

Load draught, forward, feet and inches	9-0½
aft, feet and inches	9-5½
Displacement on load draught, tons	1,824
on trial draught, tons	1,824
per inch at L.W.L., tons	22
Length over all, feet and inches	335-4
on L.W.L., feet and inches	320-0
Beam over guards, feet and inches	54-2
on L.W.L., feet and inches	40-0½
molded, feet and inches	40-0
Area L.W.L. plane, square feet	9,200
Capacity of coal bunkers, tons	200
engine-room feed tanks, tons	4
reserve feed water compartments, tons	32

MAIN ENGINE DATA.

Diameter of starboard L.P. rotor, inches	58
center H.P. rotor, inches	41
port L.P. rotor, inches	58
astern rotor, inches	47
No. of expansions, starboard rotor	8
center rotor	4
port rotor	8
astern rotor	4
Total intended brake horsepower	4,000
Expected mean speed, knots	17

SHAFTING AND BEARINGS.

Line shaft, center, steel, 106 feet long, solid, diameter, inches	6
wing, steel, 64 feet long, solid, diameter, inches	6
Propeller shaft, center, steel, 16 feet long, solid, diameter, inches	6¼
wing, steel, 36 feet long, solid, diameter, inches	6¼
Collars on shaft:	
Number	16
Thickness, inch	0½
Space between, inch	0⅞
Outside diameter, inches	10
Inside diameter, inches	6¼
Thrust shoes, number	16
effective thrust surface, square inches...top, 287; bottom,	290
Line shaft bearings: 4 starboard, 4 port, 5 center; length, each, ins.	9
Stern-tube bearings, forward and after, length, each, inches	36
After strut bearing, length, inches	36

CONDENSER AND FEED HEATER DATA.

One main surface condenser, surface, square feet	4,481
auxiliary condenser, cooling surface, square feet	140
augmenter condenser, cooling surface, square feet	135
pressure condenser (auxiliary exhaust), cooling surface, sq. ft.	321
oil cooler, condenser type, cooling surface, square feet	132

PROPELLER DATA.

Three solid Hyde manganese-bronze:	
Number of blades	3
Diameter, feet and inches	5-00
Pitch, feet and inches	4-06
Area 3 blades, square feet	8.8
helicoidal, square feet	10
Height of lower tip of blade above keel, inches	18
Immersion of upper tip of blade, inches	33

BOILER DATA.

Four three-furnace, Scotch boilers, 45-inch Morison furnaces :	
Designed working pressure, pounds	150.0
Pressure test, pounds	225.0
Ratio, G.S. to H.S.	1 to 44.3
Grates, length, feet and inches	5-8
width (total width one boiler), feet and inches	11-3
Diameter of boilers, feet and inches	14-6
Length of boilers, feet and inches	12-0¾
Grate surface one boiler, square feet	64.0
four boilers, square feet	256.0
Heating surface, one boiler, square feet	2,835.0
four boilers, square feet	11,340.0
Boiler tubes of lapwelded charcoal iron, 2¾ inches outside diameter, length, feet and inches	8-7¾
Ellis and Eaves' induced draft :	
Height of smoke pipes above grates, feet	60
Area of smoke pipe, square feet	28.0

DATA OF PUMPS AND AUXILIARY ENGINES.

		Steam × *Water* × *Stroke.*
1 Main air	Blake vert. turn beam, independent	12 × 25 × 18
1 Main circ.	Cent., two 12-in. suction, one 16-in. disch., Bath Iron Wks. Engine,	8 ... 8
2 Main feed	Blake horizontal duplex	8 × 6 × 12
4 Boiler circ	Blake horizontal duplex	4½ × 2¾ × 4
2 Oil pumps	Blake horizontal duplex	6 × 5¾ × 6
1 Donkey F. & B.	Blake horizontal duplex	10 × 8½ × 12
1 Sanitary	Blake horizontal duplex	6 × 5¾ × 7
1 Aux. cond	Blake combined air and circ.	5½ × 6 × 7
1 Donkey feed.	Blake horizontal duplex	4½ × 2¾ × 4
1 Fresh water	Blake horizontal simplex	3½ × 3½ × 4

2 Dynamo engines, 35 kw., General Electric, turbo generators, Curtis.

2 Forced-draft blowers, 84-inch diameter fans, 17-inch face, 5 inch × 5 inch, Bath Iron Works Engine.

On the morning of May 23, 1907, at 9 o'clock, the *Camden* left the dock at the Bath Iron Works, Bath, Me., and proceeded to the trial grounds at Boothbay Harbor, Me. Four standardization runs were held over the measured mile, under natural draft, followed by two runs under forced draft. At the conclusion of these runs a two-hour run was held under forced draft. The vessel then returned to Bath, Me., and secured at the dock at 3·30 P. M. During the first hour of this latter run a grinding noise was heard for several seconds.

STANDARDIZATION TRIALS OF S. S. *CAMDEN* ON COURSE OFF SOUTHPORT, ME., ON MAY 23, 1907.

Number and direction of run.	Elapsed time.	Steam pressure.				Vacuum, in inches.	Auxiliary exhaust, pounds.	Revolutions.				Torsion meter readings.			Temperature, degrees Fahr.				Speed, in knots.	Shaft horsepowers.			
		At engine ab. atmosphere.	Center turbine.	Starboard turbine.	Port turbine.			Starboard shaft.	Center shaft.	Port shaft.	Mean 3 shafts.	S	C	P	Injection.	Discharge.	Hot well.	Feed.		Starboard shaft.	Center shaft.	Port shaft.	Total.
1 N	4.556	142	35	*3.0	*4.0	28.2	1.5	357.7	346.8	357.7	354.1	.28	.34	.33	43	60	68	193	13.17	367	436	432	1235
2 S	4.193	143	55	*3.0	†1.0	28.2	1.5	410.2	395.9	407.8	404.6	.36	.41	.38	44	66	68	190	14.31	544	600	568	1712
3 N	3.646	137	79	†7.6	7.3	27.9	1.0	466.3	452.5	463.5	460.8	.48	.50	.48	42	68	68	190	16.46	811	837	815	2463
4 S	3.533	134	100	11.0	12.0	27.6	2.5	495.3	481.2	498.2	491.6	.58	.60	.58	42	74	74	178	16.98	1053	1068	1059	3180
5 N	3.123	152	139	17.7	20.0	27.7	3.0	563.5	544.3	560.4	556.1	.72	.76	.74	42	74	74	174	19.21	1487	1532	1521	4540
6 S	3.190	152	139	18.0	20.0	28.0	3.0	558.0	542.3	561.1	553.8	.71	.78	.73	42	74	74	170	18.81	1447	1566	1502	4515

*Inches vacuum. †Pounds.

Run.	Counter Readings.									Run.	Auxiliaries, Double strokes.				Revs. circ. pump.
	Starboard.			Center.			Port.				Air.	Feed	Oil.	Sanitary	
	On.	Off.	Diff.	On.	Off.	Diff.	On.	Off.	Diff.						
1 N	25,000	26,630	1,630	32,230	33,810	1,580	51,610	53,240	1,630	1 N	50	22	19	24	192
2 S	30,670	32,390	1,720	37,750	39,410	1,660	57,210	58,920	1,710	2 S	52	22	20	24	192
3 N	36,470	38,170	1,700	43,470	45,120	1,650	63,030	64,720	1,690	3 N	52	24	19	24	188
4 S	42,730	44,480	1,750	49,540	51,240	1,700	69,240	71,000	1,760	4 S	48	26	20	24	186
5 N	48,550	50,310	1,760	55,260	56,960	1,700	75,130	76,880	1,750	5 N	52	34	20	24	192
6 S	54,950	56,730	1,780	61,430	63,160	1,730	81,500	83,290	1,790	6 S	52	32	19	24	186

$$\text{H.P.} = \frac{T \times R \times 19{,}455}{5{,}255}$$

for center shaft.

$$\text{H.P.} = \frac{T \times R \times 19{,}270}{5{,}255}$$

for wing shafts.

T=torsion meter reading.

R=revs. per min.

19,455 and 19,270 are calibration constants.

U. S. S. *CAMDEN*, COURSE OFF SOUTHPORT, ME., ON MAY 23, 1907.

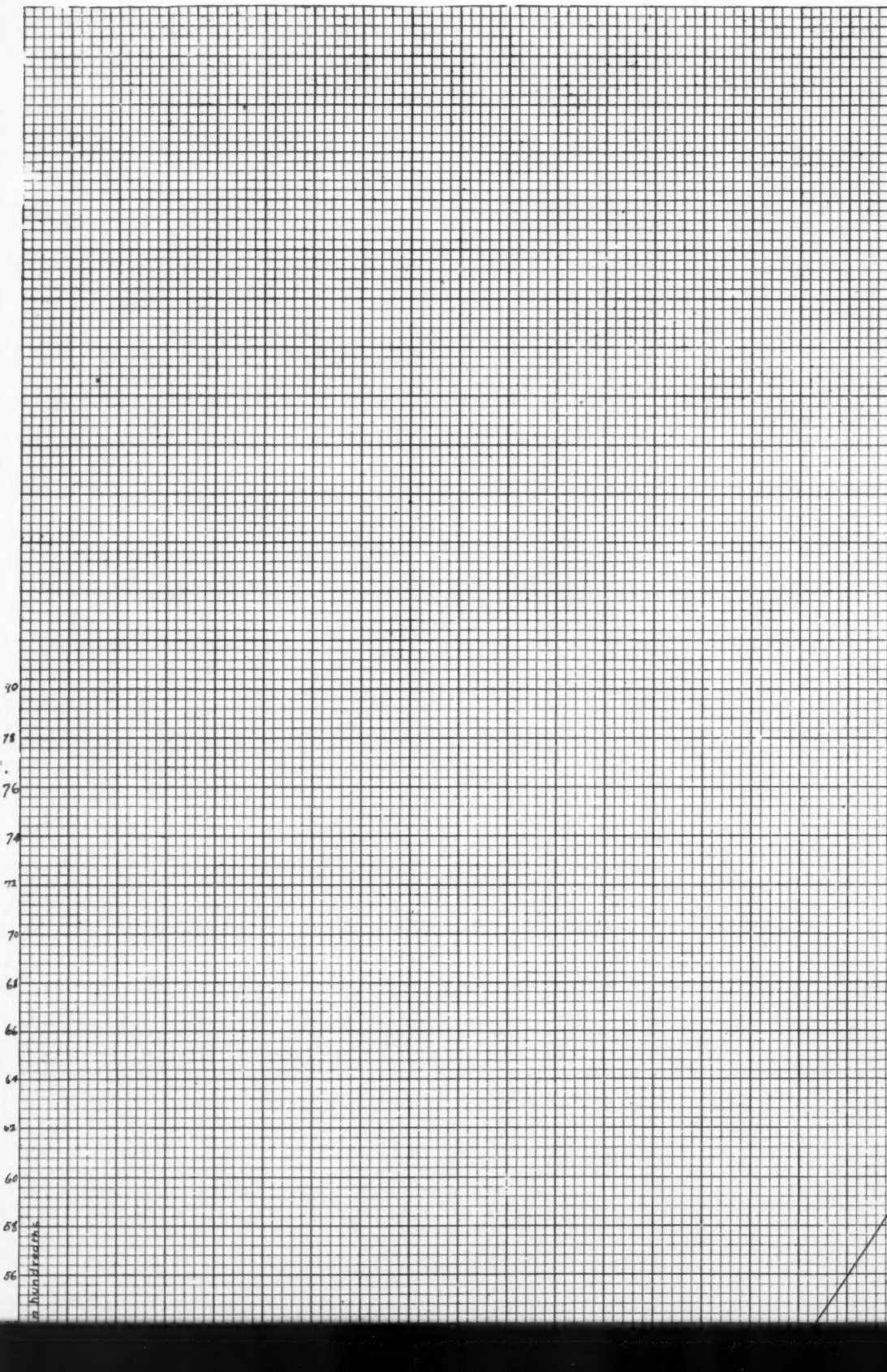
TAKEN FROM CURVES.

Speed, knots.	Revolutions per minute—mean of three shafts.	Steam pressure, H.P. turbine.	Total shaft H. P.—three shafts.
5	128	8	200
9	240	18	575
13	357	40	1,275
15	419	61	1,850
17	485	95	2,855
19	555	144	4,575

DATA TAKEN DURING LAST HOUR OF TWO-HOUR SEA RUN.

Time.		Auxiliaries, dbl. strokes.					Steam at turbine.				Temperatures.					Counter readings.			Revs. per minute.			Torsion meter readings.			Shaft horse-powers.			
H.	M.	Air pump.	Feed pump.	Oil.	Circulating.	Boiler steam.	Starboard.	Center.	Port.	Vacuum.	Injection.	Discharge.	Hot well.	Feed.	Aux. exhaust, lbs.	Starboard.	Center.	Port.	Starboard.	Center.	Port.	Starboard.	Center.	Port.	Starboard.	Center.	Port.	Totals.
1	00	47	34	20	190	149	16.5	140	19.5	27.6	43	75	75	180	1.5	632,410	609,930	606,260	...	...	...	...	...	...	...	...	...	...
1	10	46	29	20	190	148	16.5	137	19.3	27.9	43	75	73	176	2.0	638,210	615,500	612,050	580	557	579	.70	.76	.70	1,488	1,567	1,485	4,540
1	20	45	31	20	188	142	15.5	132	18.1	27.8	43	74	72	176	2.0	643,720	620,790	617,510	551	529	546	.67	.74	.67	1,353	1,449	1,341	4,143
1	30	45	30	14	180	145	15.9	134	18.8	27.9	43	76	73	178	3.5	649,280	626,050	623,040	556	526	553	.68	.76	.68	1,386	1,480	1,379	4,245
1	40	44	28	20	174	142	15.4	132	18.2	27.9	43	78	74	178	3.0	654,870	631,410	628,610	559	536	557	.71	.76	.68	1,455	1,508	1,388	4,351
1	50	44	28	21	168	137	14.0	126	16.5	27.6	43	75	73	180	3.0	660,280	636,600	634,010	541	519	54C	.66	.72	.64	1,309	1,383	1,267	3,959
2	00	37	28	20	168	137	15.0	126	17.1	27.3	43	75	73	180	1.5	665,750	641,800	639,360	547	520	535	.66	.72	.62	1,323	1,386	1,216	3,925
Avg...		44	29	20	180	143	15.5	132	18.2	27.7	43	75	73	178	2.4	...	...	...	556	531	552	.68	.743	.665	1,385	1,463	1,346	4,194*

*Speed corresponding, 18.6 knots.


80
78
76
74
72
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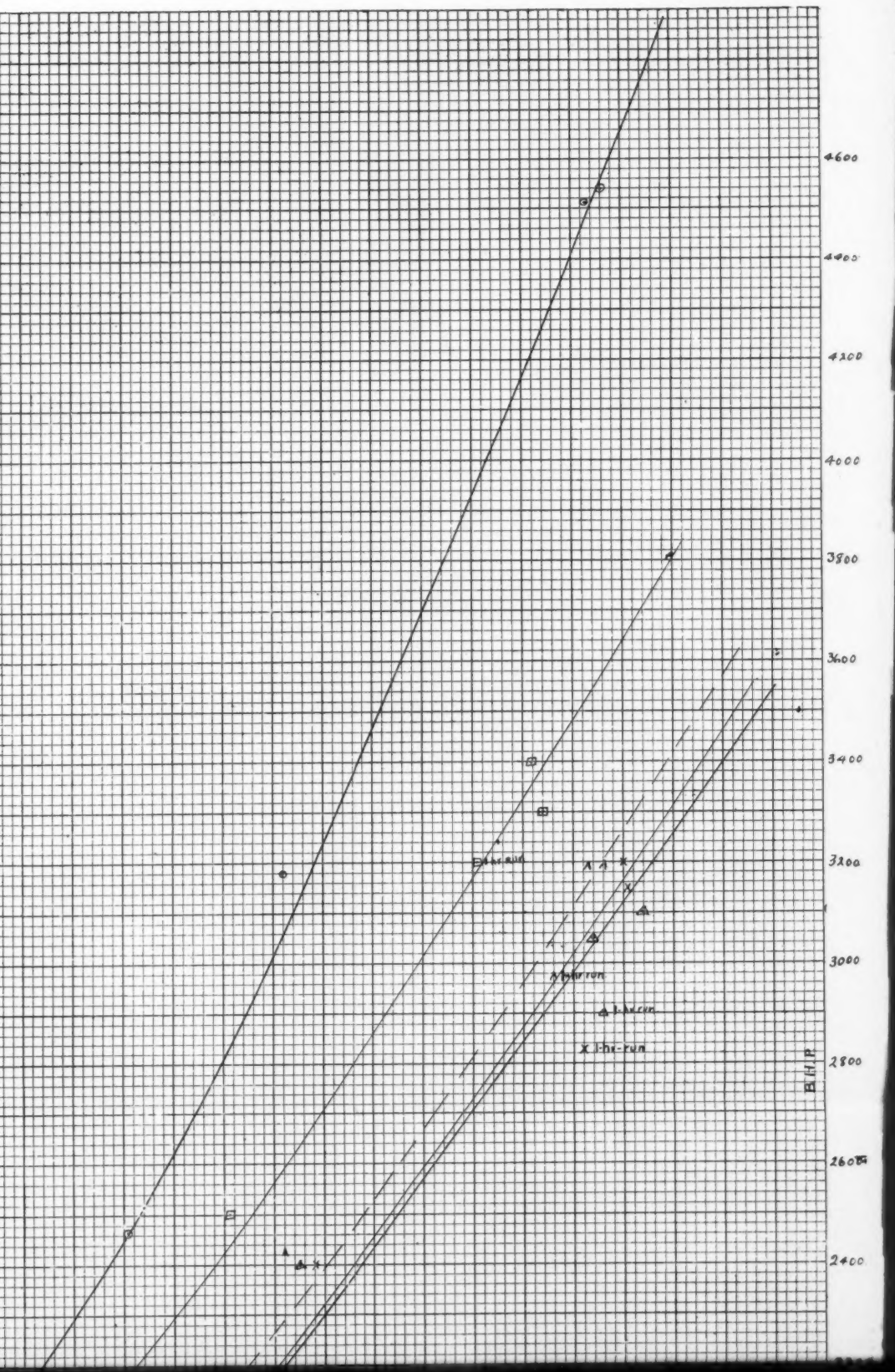

4600
4400
4200
4000
3800
3600
3400
3200
3000
2800
2600
2400
B.H.P.
Morrison
1-hr. run
1-hr. run

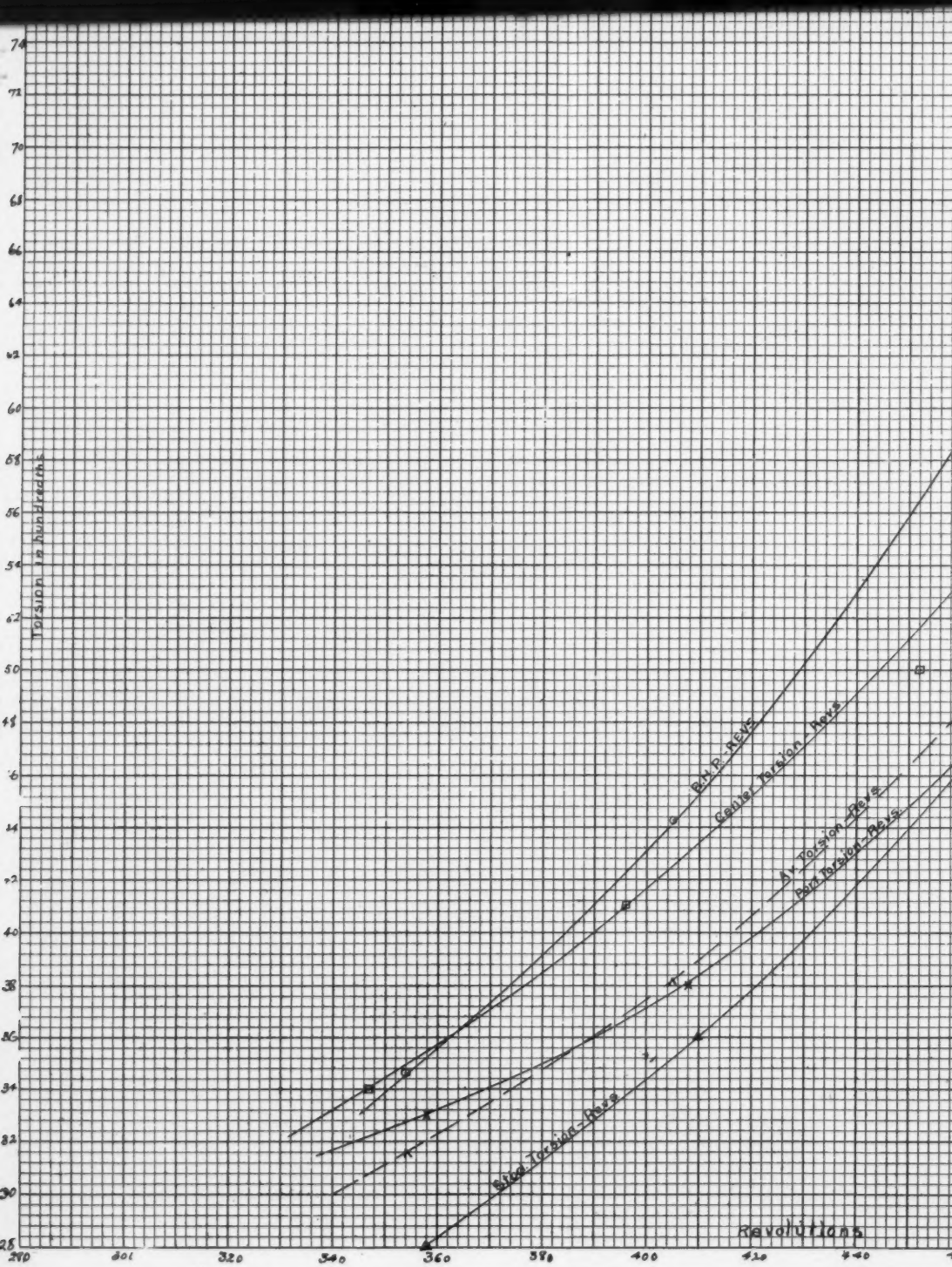
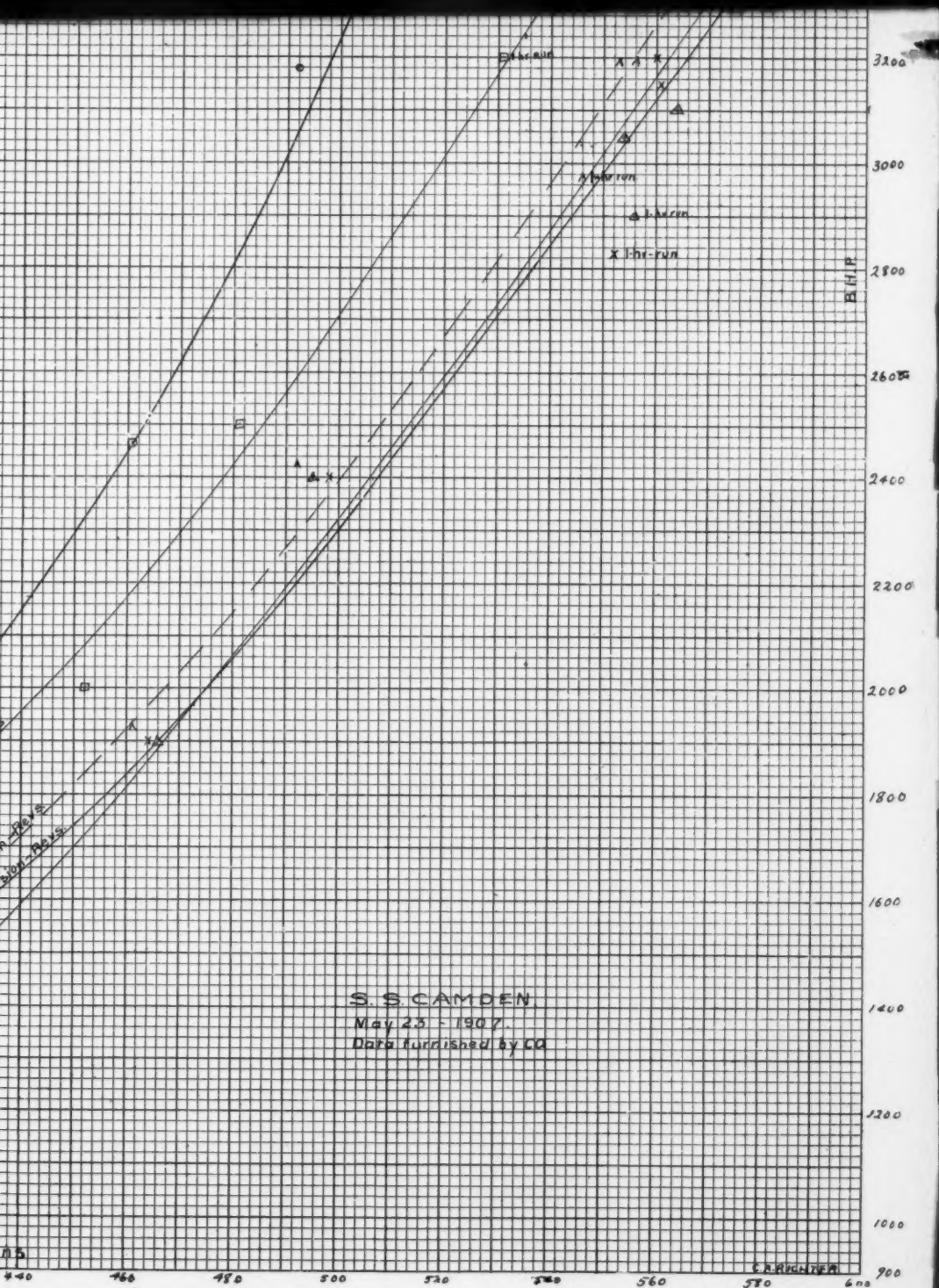


Plate III.—S. S. "CAMDEN," REVOLUTION



REVOLUTION AND HORSEPOWER CURVES.

This was attributed to the astern dummy in the port low-pressure turbine. The turbines were slowed down and worked up again gradually and the noise ceased. The turbines continued to run smoothly during the remainder of the run, and it was concluded that no injury had resulted. This conclusion was borne out several days later when the turbine casing was lifted and a thorough examination made. Aside from this occurrence, the machinery worked most satisfactorily throughout all of the runs. No vibration was noticed in the shaft alley.

The two-hour run was primarily for determining the coal consumption, but, owing to the fact that the turbines had to be slowed down, the data secured were unreliable and are not noted herein. The brake horsepower was determined through the use of a Denny & Johnson torsion meter.

The data were secured during the vessel's trials.

Plate No. 3 shows a revolution and horsepower curve prepared from the above data.

At the conclusion of the trial the telegraph was thrown from full ahead to full astern. The time consumed in operating the turbines was inappreciable. The vessel came to a dead stop in two minutes and thirty-four seconds.

MODERN ARMOR AND ARMOR-PIERCING PROJECTILES.

By Lieut. H. J. Jones, A. R. C. Sc. (Lond.), A. O. D.,
Inspector of Ordnance Machinery.

The problem of armor and armor-piercing projectiles is one of the most interesting and complex in the study of ballistics. Its peculiar difficulty lies in the fact that information concerning the curious phenomena that have from time to time been recorded has been information of a more or less qualitative kind, obtained under conditions of great dissimilarity, when the observation of the particular facts recorded was not the object for which the experiments were primarily devised. Organized trials of standard plates under standard conditions would involve such expense that private firms can hardly be expected to undertake them, and, as a consequence, most of our information concerning the behavior of armor under attack has been obtained from proof trials for acceptance or premium, and, naturally, under these circumstances the commercial aspect has predominated.

It thus comes about that our knowledge of this important branch of ballistics is confined to general comprehensive statements which simply indicate the tendency of a series of isolated observations. So long as the materials and processes are more or less secret and confidential, and the value of a plate is determined by its ability to cope with a special class of projectiles, so long must any solution of the problem of armor and projectiles be merely tentative and unsatisfactory. Nevertheless, there are certain aspects of the subject of which it may be useful to define the general scope and character, although from the gunner's point of view they may not appear to be intrinsically the most significant.

ARMOR PLATES.

Modern armor is universally made of one or other of the ferro-alloys, usually nickel-chrome, low in carbon, with a small percentage of manganese, tungsten, &c.; the proportions are usually considered a trade secret, and in all probability vary with the thickness of the plate. A good average Krupp plate contains $3\frac{1}{2}$ per cent. nickel, $1\frac{1}{2}$ per cent. chromium, $\frac{3}{10}$ per cent. carbon, and $\frac{7}{10}$ per cent. manganese. The plates may be cast, up to 6 inches or 7 inches in thickness, or forged; and they may be cemented or non-cemented. Cementation results from a superficial carburization, which enables a glass-hard face to be obtained without in any way affecting the tough qualities of the remainder of the plate. This effect is produced by subjecting the face to a temperature suitable for the solution of carbon placed in contact with it, the subsequent tempering permitting the production of a surface of any degree of hardness. The depth to which the cementation has effect depends upon the composition of the plate, its temperature, the time of exposure and the nature of the carburizing process. The solid carbon process of Harvey, which gave a face composition of as much as $\frac{7}{10}$ to $\frac{8}{10}$ per cent. of carbon, consisted in covering the furnace base by a few inches of sand before the plate was laid on it. The upper face of the plate was then covered with a mixture of animal and vegetable charcoal about a foot in thickness, over which a covering of tiles was put, the doors of the furnace being then bricked up and a high temperature maintained for many days.

The modern method of Krupp consists in keeping ordinary illuminating gas or petroleum vapor under pressure in contact with the heated plate, a process that was covered by Thwaite's English patent in 1893, and said to possess the advantages of more uniform and definite action, combined with lower temperature, shorter time and reduced cost.[1] In general the cementing materials belong to three classes—those acting by the formation of carbonic oxide, those acting by means of a

[1] Castner. "Stahl und Eisen," September, 1895.
Edwards. "Armor Plate." British Association, August, 1906.

cyanide and those employing hydrocarbons. Experiments made with pure carbon in a vacuum have demonstrated that no cementation takes place unless air is present and carbonic oxide formed. In the case of the cyanide there is simple decomposition, in which the carbon is liberated to be dissolved in the steel; while, when hydrocarbons are employed in the form of illuminating gas, petroleum vapor, &c., the carbon is released by dissociation.[2] The chemical and structural changes effected by cementation have been studied by Arnold;[3] and Charpy has published a most complete account of some quantitative researches on the cementation of steel with different carburizing materials in electric furnaces at definite temperatures.[4]

Cementation seems to be at its best for about a 6-inch plate, the skin effect, after the differential heat treatment, extending to about 2 inches, and the 2 to 1 proportion between tough back and hard face represents good practice. It has been found necessary that the steel for cemented and water-hardened face armor must be much more carefully selected and treated than was that for the early chilled-steel armor; and it is eminently necessary that the plate should gradually grade from the face quality to the back quality, otherwise under attack the hard skin flakes off or cracks in much the same way as did the old compound plates. Most makers object to the cementation of thin plates, the general experience being that, unless the plate is small and flat, cementation is not a success, warping and cracking of the plate being unavoidable. Krupp, however, professes to have solved the problem of manufacturing thin curved plates possessing all the features of Kruppized armor, as they are used, for instance, in the turret bases of cruisers.[5] Below 4-inch or 5-inch, which is usually considered to be the limit of cemented armor, the

[2]Guillet. "La Cementation des Aciers au Carbone et des Aciers Speciaux." Societé des Ingenieurs Civils de France, February, 1904.
"Nouvelles Recherches sur la Cementation des Aciers au Carbone et des Aciers Speciaux." "Comptes Rendus," June, 1904.
[3]Arnold. "Micro-chemistry of Cementation." "Proceedings," Iron and Steel Institute, August, 1898.
"Diffusion of Elements in Iron." "Proceedings," Iron and Steel Institute, May, 1899.
[4]Charpy. "Sur la Cementation du Fer." "Comptes Rendus," April, 1903.
[5]Krupp. "The Engineer," January 31st, 1902.

Charpy high-nickel cemented plates appear to be the best, as they can be made down to about 1 inch in thickness, and are thus of great value for destroyers and the shields of light Q.F. guns.[6] Usually the 5-inch limit of cemented plates marks the limit for non-cemented plates, the treatment of which is identical with that of the ordinary cemented plate, excepting that there is no superficial treatment, and the plate retains a tough quality throughout. We appear to be the only people to make non-cemented armor for ordinary purposes. Within recent years both Krupp and Hadfield have successfully produced cast-steel armor up to 6-inch and 7-inch in thickness, with a factor of merit approaching 2; this nature of armor seems particularly suitable for ammunition tubes, gun shields, &c.

The manufacture of a modern cemented plate involves a large number of costly processes, and a series of special heat treatments.[7] The steel ingots weighing from 60 to 100 tons are made in open-hearth furnaces, with either acid or basic linings, and chemical analyses are made to ensure the composition of the steel being within the limits laid down by practice. The most important point in the subsequent treatment of the ingot is the limited range of temperature at which it is possible to get satisfactory results, for the nickel-chrome alloy as forged may, by suitable heat treatment, have its tenacity increased from 40 to 90 per cent., and its ductility from 8 to 30 per cent., &c.[8] For armor plate reheating the temperature must be known to within 10 degrees or 20 degrees centigrade, and one of the simplest foundry methods of temperature measurement by an iron ball and water calorimeter appears to afford sufficiently accurate indications for practical purposes.[9] For a comparison of the value of various pyrometric instruments for industrial purposes reference may be made to the works of Le Chatelier, to a discussion at the Iron and Steel Institute,[10] and to a paper by Whipple.[11]

[6] Tresidder. "Brassey's Annual," 1905. Pp. 366–371.
[7] Johns. "The Production and Thermal Treatment of Steel in Large Masses." "Proceedings," Iron and Steel Institute, 1903.
[8] Hadfield. Presidential Address. "Proceedings," Iron and Steel Institute, 1905.
[9] Gledhill. "Proceedings," Iron and Steel Institute, May, 1904.
[10] Exhibition of Pyrometers. "Proceedings," Iron and Steel Institute, May, 1904.
[11] Whipple. "Pyrometrics for Industrial Purposes." Cleveland Institution of Engineers, December 5th, 1904.

The steel ingot is left in the molds, but is not allowed to get cold, it being removed to a forge, and heated by gas for many hours. It is then forged, sometimes at the rolling mills, sometimes under a press, and sometimes partly at the rolls and partly at the press. The correct temperature for working the metal at the rolling mills is secured by the use of pyrometers; micro tests and cooling curves are made, and the critical points determined in the laboratory. When roughly forged the ingot goes back to the reheating furnace for several hours, after which it is rolled roughly to dimensions. The plate is then supported on knife edges, some feet above the ground, so as to admit of a free supply of air, and when the plate has cooled to a certain temperature it is put into a special low-temperature furnace, after which it is softened by immersion in water. Pneumatic scaling hammers then remove the scale and the plate is machined, the surface being carefully examined for cracks. The plate is then taken to the cementing furnace, and when in pairs the carburizing medium is sandwiched between them. The time during which the plate is undergoing cementation varies from two to three weeks, and pyrometric records are either taken at frequent intervals of about 15 minutes, or an automatic arrangement is used which rings a bell when the temperature falls too low. When the process is complete the gas is shut off and the furnace allowed to cool for another week, after which the plate is reheated and plunged into rape or other oil to harden, the time of the last reheat being determined by the size, weight and thickness of the plate. The plate is again heated and plunged into a circulating cooling solution, after which it goes to the presses and is bent to the molds, the latter having been taken directly from the ship, or, in the latest practice, from the ship's drawings.[12] The plate is again heated, immersed, and allowed to cool. It is then sent to the machine shops and finished to size. When the final differential heat treatment of the cemented plate has taken place, the face is so hard that, without special local treatment, holes

[12] White. "Modern Warship." Cantor Lectures. Society of Arts, 1906.

cannot be drilled or tapped. To overcome this difficulty, it has been usual either to protect the surface of the plate during cementation in places where holes were likely to be required, or else to make accurate drawings and patterns, and to do all work on the plate before the final hardening. The first method is attended by the difficulties arising from errors in the foundry and the drawing office, and also that no certain means of protecting patches of the plate has been discovered. The second method, although slow and costly, is common in this country, the plate being bored, tapped and countersunk, and the holes filled with clay, before the plate is hardened. Attempts have also been made locally to anneal the plate by means of the blow pipe or electric arc, but without conspicuous success. In America, and to some extent in this country and Germany, electric annealing currents are passed through the plate by pressing two copper contacts, cooled by circulating water, against the plate, and this method has also had successful application to the softening of strips of metal for the removal of slots for sights.[13]

The plate, having been finally prepared, now undergoes what is known as the differential heat treatment, in which the face is raised to a higher temperature than the back, so that, whilst on sudden cooling the face is rendered glass-hard, the back retains its essential characteristics of toughness. The plate then goes back to the machine shops, and the surface curves are tested to see whether they correspond with the molds. If the plate has warped it is reheated, and the differential heat treatment repeated after the requisite bending. As a rule, it is possible slightly to flatten hardened cemented plates of moderate thickness by compression of the face, but bending, which puts the face in tension, almost always results in cracking. Any work that has to be done on the plate after the final heat is performed by grinding or disc cutting. Bending and tensile tests are usually carried out on specimens cut from the plate, and in the best practice some modification of Brinell's impression test, the Borba and Frémont drop test,

[13] Semp. "Local Annealing of Hard-Faced Armor Plates." American Institution of Electrical Engineers, 1896.

and vibratory tests on a duplex vibratory machine are also made.

The manufacture of cast armor is now in a highly efficient state. The metal is cast in a mold so formed as to produce a plate with a face having ridges and channels. The hot casting is removed to a special furnace, and allowed to cool slowly to prevent initial strains, after which the sand is removed and the casting carefully cleaned. It is then placed in a furnace, and upon the corrugated surface a carbonaceous composition is placed to a thickness of several inches, the furnace being raised to a temperature between 900 degrees and 1,100 degrees centigrade, which is maintained for several days. The plate is then allowed gradually to cool. The plate is again reheated and cooled four or five times, with a gradual reduction of the temperature to which it is reheated, and at the final heating the plate is dipped or sprayed with oil or water, according to the hardness required, so that the corrugated face, being hotter than the back, will become hard, while the back remains soft and tough.[14]

The fitting and erection of naval armor involves several operations of considerable technical interest. As an illustration of modern practice we may take a recent "first-class" battleship, the typical particulars of the armor and armament being as follows:[15]

ARMOR.

Belt, inches,	9
Deck, inches,	2—1
Side above belt, inches,	8—7
Bulkhead, inches,	12
Heavy-gun positions, inches,	12—6
Secondary-gun positions, inches,	7

ARMAMENT.

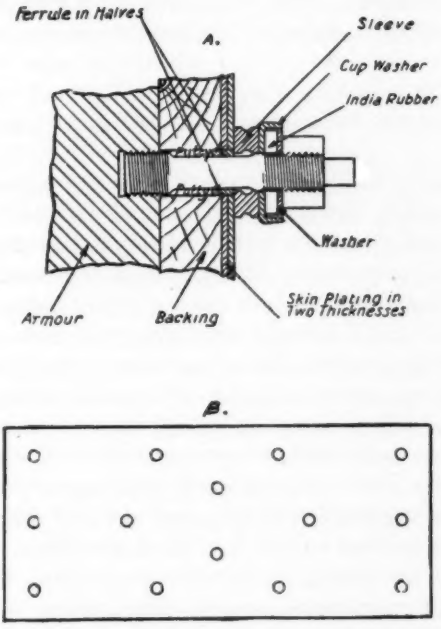
Four 12-inch, four 9.2-inch, ten 6-inch, twenty-eight small guns.

The latest practice is to carry the side armor throughout the length of the ship, but to reduce the thickness to 2-inch

[14] Hadfield British Patent No. 17,174, 1905. "The Engineer," "Armor Plates," January 5th, 1906.

[15] "Brassey's Annual," 1905, page 235.

on bows and stern, this thin armor being riveted directly to plating and carrying no backing behind it. The armor bolts are of mild steel, Fig. 1, the number used being one bolt for



A = Section through armor bolt.
B = Disposition of bolts in plate, 14½ feet by 6½ feet.

Fig. 1.—ATTACHMENT OF ARMOR TO SHIPS.

every 7 square feet of plate. The usual dimensions, &c., of armor bolts are as follows:

Plate.	480 lbs.	400 lbs.	360 lbs.	320 lbs.	280 lbs.	240 lbs.	200 lbs.	160 lbs.
Diameter of shoulder of bolt at cup washer.....	4	3⅞	3½	3¼	3	2⅞	2½	2¼
Depth of hole in plate..	3⅞	3 9/16	3¼	2 13/16	2 11/16	2⅝	2 1/16	1⅝
Depth of bolt in plate.....	3¼	3	2⅝	2⅜	2¼	2	1⅝	1⅜

N.B.—The weight of one square foot of armor plate, 1-inch in thickness, is taken to be 40 pounds.

The bolts are made watertight where they pass through the skin plating, and are reduced in the center of their length to increase the resilience and to render them less liable to break under impact. The sleeves are fitted merely to fill up the length of the bolt before placing the nut on, and care is taken to get the lower row of side-armor bolts as near the lower edge as possible. The armor itself is not made watertight, and water finds its way between the armor and the backing.

The backing is East Indian teak, worked longitudinally on the ship's side. Behind the 360-pound armor it is 4 inches thick and tapers to about 2 inches behind the thinner armor.

The backing is worked in such thicknesses that the outer surface of the armor is flush, viz: 4 inches behind the 360-pound armor and 6 inches behind the 280-pound armor.

The plating is in two thicknesses of 17½-pound plate behind the 360-pound, 320-pound and 280-pound armor, and two thicknesses of 15-pound plate behind the 200-pound and 160-pound armor, each thickness serving as butt straps and edge strips to the other. In the most recent practice only one thickness of 30-pound plating is used, the butt straps and edge strips being worked on the outside of the ship. The edges and butts of the plating are planed and accurately fitted, being riveted with countersunk points. The plating is connected to the middle deck by a single angle bar 4½ by 4½ by 18 pounds, and to the main deck by a 4 by 4 by 13-pound angle.

The frames behind the heavy armor consist of two angle bars 10 by 3½ by 20-pound, and behind the thin armor the frames are 10 by 3½ by 24-pound down to 6 by 3 by 15-pound. The heavy frames are spaced about 2-foot centers, and the others at 3-foot centers. Two horizontal girders are worked between the frames intercostally. The plating is made watertight by calking the edges and butts, and the faying surfaces are coated with red-lead paint. When armor-bolt holes are cut out, the edges between the two plates are also calked, every effort being made to ensure the plating being watertight, irrespective of the armor and the backing. Red lead

is injected by means of a force pump between the backing and the plating. The outside of the backing and the inside of the armor is thickly coated with a stiff mixture of white and red lead before the plate is put on, and after the plate has been secured a thin mixture of the same materials is then injected.

Armor for coast-defence ordnance has quite a different character to that for naval purposes. Land-service shields usually have curved inclined fronts, and are supported by curved stays bolted to the sides of the carriage. Considering the small clearances allowed, these stays do not appear to possess adequate stiffness to prevent the shield setting back on impact and jamming the gears, particularly so when we remember that high-sighted guns are always under oblique attack, and that under these circumstances the work of impact is mainly the work of distortion or displacement of the shield. This aspect of the matter, with special reference to the box-like structure of naval casemates, seems only to have been considered in the French service. The shields contain openings for the gun and sights, and are either in one piece, or in two pieces let into one another at the front and bolted together by a thick plate on the inside.

ARMOR-PIERCING PROJECTILES.

Modern armor-piercing projectiles are made from a nickel-chrome ferro alloy, and are either cast or forged. Hadfield has given the following proportions of elements, other than iron, as being typical for air-hardened projectiles. Carbon, .8 per cent.; silica, .2 per cent.; sulphur, .04 per cent.; phosphorus, .04 per cent.; manganese, .12 per cent.; nickel, 2 per cent.; and chromium, 2 per cent.[16] The ingots for forged armor-piercing projectiles are usually made so that two projectiles can be obtained from one ingot, and the treatment of the ingot is somewhat similar to that adopted for the ordinary nickel-chrome plate. The ingots are not allowed to cool, but

[16] Hadfield. British Patent No. 6089. 1901.

are reheated and forged hydraulically. The forging is reduced roughly to finished size, the points being formed between swages, after which the projectiles are slowly reheated and annealed.[17] Armor-piercing shells have their cavities formed by punches and dies, and in the case of shells, such as the American, the base is closed by a plug screwed into the walls, which are not upset. In our service the base is continuous with the walls, and is closed by a base plug, the internal cavity being first punched or pressed, and the walls upset in a die when the ends only are hot. The cast-steel projectiles are cast point downwards, the body and dead head being in a sand mold, and the nose, or part of it, in a cast-iron chill. The mold is first dried and the chill coated with a carbonaceous composition, the core being carried on a spindle in the usual way. After casting, the shells are placed in an annealing oven, at about 1,400 degrees Fahrenheit, for a day or so, several castings being annealed at the same time. The shells are then machined, the dead head cut off and the cavity trimmed, the ogival head being either cut or ground to the correct radius. In our service the radius of the head is 2 calibers, corresponding to a nose angle of $2 \tan^{-1} \frac{\sqrt{7}}{3}$, or 83 degrees—12 minutes for all shot and shell. In America the 12-inch A.P. shot has a radius of head 24.18 inches, the 12-inch light shell 18.00 inches and the 12-inch heavy shell 18.075 inches.[18] A similar variation in practice obtains in other countries.

(In all probability this matter needs revision. The original series of experiments to determine the best shape of head for this class of projectile were carried out at the low striking velocity 1,500-foot seconds, against unbacked wrought-iron 14—12-inch plates, and unbacked 12—10-inch compound plates.[19] In each case the order of merit was assigned according to the depth of the indent made, a method which would be entirely unsuitable for modern hard-faced Krupp or

[17] Carnegie. "Armor-piercing Projectiles." "Proceedings," I. C. E., February, 1903.
[18] "The Engineer," "American Armor-piercing Projectiles," November 19th, 1897.
[19] Cunhill, "Experiments with Armor-piercing Projectiles," "Proceedings," R. A. Inst., Vol. XI, No. 7.

Terni plates. From the results of experiments on hardened steel plates by dropping heavily-loaded punches on them, I find the more highly elastic the plate the more blunt should be the nose, particularly for oblique impact. Furthermore, when the plate is defeated by what is generally called "boring action," I find the diameter of the punch relative to the thickness of the plate to be a deciding factor; whereas, when the plate is defeated by a smashing action, I find the angle of the nose and the diameter of the punch to be unimportant, the velocity rather than the weight appearing to decide the result; that is to say, a certain increase in the weight of the punch is less effective than the same percentage increase in the striking velocity.)

The head having been brought to the correct radius the base is then trimmed, bored and screwed, the groove for the driving band turned, and the projectile made ready to receive its cap. It is then brought to the correct weight, examined and hardened, after which it is ground to gauge, if necessary.

The method adopted for hardening projectiles varies in much the same way as the method for plates. Besides heat treatment, even cementation of the head has been carried out, the nose being afterwards tempered by cooling in oil. One method of hardening consists in heating the projectile, suspended point downwards, in a furnace, the point and shoulder being raised to a hardening temperature, whilst the point itself is protected from the direct action of the heat. The projectile is then removed to a hardening tank, and a cooling medium circulated under high pressure throughout the cavity, the projectile being afterwards completely immersed.[20] At Terre Noire the projectiles are oil hardened, the quality of the steel differing in different shells, being hardest in the largest calibers. The St. Chamond projectiles, generally of crucible steel, are heated to a cherry red, and gradually plunged into an oil bath until cold; but in some cases the projectile is plunged, for about eight or ten minutes, only up to the front band, after which it is completely immersed until

[20] Hadfield, American Patent, May 27th, 1898.

cold.[21] Armor-piercing projectiles manufactured at the Toubia Arsenal used to be heated in an upright gas furnace to a temperature between 780 degrees and 800 degrees centigrade; they were then transferred to an iron tank, and a jet of oil under a pressure of half an atmosphere acted on the point of the projectile until the tank was full, the capacity of the tank being sufficient to hold a weight of oil equal to ten times the weight of the projectiles. The method subsequently adopted for quick-firing armor-piercing projectiles was to heat them to about 740 degrees centigrade, and then plunge them into a bath of oil at about 24 degrees centigrade. They were then hardened again at the point, a jet of water being played on the inside and outside. The caps were also hardened, the top being heated to a bright red heat and the cap then plunged into water.[22] Air hardening, as introduced by Hadfield, is effected by directing a cold blast on the inside and outside of the projectile, which is raised to a higher temperature than when cooled by oil or water.

To test for dangerous initial strains the shells are kept for about three months, if possible, and some of them, after having been immersed in hot water up to the driving band, are plunged into cold water at least 130 degrees Fahrenheit lower than the temperature of the hot water. In America, before acceptance for ballistic tests, the projectiles are cooled to about 40 degrees Fahrenheit, and then suddenly plunged into a bath of water at from 180 degrees to 212 degrees Fahrenheit. When thoroughly heated, each projectile is plunged, with its axis horizontal, half-way into a bath of water at a temperature not greater than 40 degrees Fahrenheit, and after a brief period the projectile is turned over for a like immersion of the opposite side. The projectile is then removed from the bath, and tested under a hydraulic pressure of 500 pounds per square inch.[23]

With regard to the design of armor-piercing projectiles,

[21] Nazro, "Modern Mechanism." Macmillan & Co., 1899.
[22] Cubillo, "Armor-piercing Projectiles" (Correspondence). "Proceedings," I. C. E., February, 1903.
[23] "The Engineer." "American-Armor-piercing Projectiles," November 19th, 1897.

the general practice is to put no restriction either on the material or the method of manufacture, the projectiles being accepted solely on the ballistic and other tests. The weight, form of head, maximum length, diameter over band, size and shape of driving band, are usually specified for armor-piercing shot; but shape, size, position of cavity and means of closing the base, are left to the manufacturer. In the case of armor-piercing shells the minimum capacity and minimum thickness of walls are also supplied.

A method by which the thickness of the base of a shell can be checked has recently been published, but it rests on the altogether artificial assumption that the base of the shell will tend to fail by shear along a cylindrical section parallel with the walls.[24] This, of course, the base will not do. The minimum thickness of the base to carry the powder pressure can be determined on the assumption that the base will behave as a plate, fixed at its edges, under a uniformly distributed load; then d being the diameter of the shot, p the maximum pressure on the base, and f the safe tensile stress, in the usual units, the thickness t is given by

$$t = \frac{d}{3}\sqrt{\frac{2p}{3f}}.$$

When the base is closed by a screwed plug it has to be ensured that the load on the plug can be carried by the shear strength of the thread. The usual thickness of base is considerably greater than that necessary to prevent fracture on firing; actual bases probably have a factor of safety of from 5 to 7.

The minimum thickness of the walls of an armor-piercing shell to prevent "setting-up" in the gun is much less than that necessary to prevent "setting-up" when striking a plate. The former can be calculated, but there is absolutely no theoretical guidance applicable to the determination of the latter. The minimum thickness of our armor-piercing shells closely follows a rule—

$$t = .144d + .084,$$

[24] Freeth. "Journal of the R. A.," December, 1905.

d being the caliber and t the minimum thickness in inches. American shells follow a rule—

$$t = .19d + .03,$$

and from the particulars available of French and German shells, the general practice appears to be to make t from one-fifth to one-seventh the caliber.

The simplest way to design a shell when the caliber, thickness of base, and minimum thickness of the walls has been decided, is to draw a section of the shell to a conveniently large scale, preferably on squared paper, and, assuming a probable shape of cavity, to determine graphically the weight of the shell, capacity of the cavity and position of center of gravity. A little experience enables one to adjust the shape of the cavity to bring the shell within the specified limits of weight, capacity, &c.

The factors affecting weight and proportion of projectiles do not generally permit of exact definition, but as a working rule we may take it that for service armor-piercing projectiles, the weight in pounds and the caliber in inches are connected by a rule, such as—

$$3w = d^{3.172}.$$

The tests of armor-piercing projectiles consist in firing them at certain velocities against a specified type of plate, the target being sometimes completed by a suitable thickness of backing and one or more skin plates. In the case of American armor-piercing shots, two are fired normally at a $1\frac{1}{8}$-caliber plate, with velocity sufficient to carry them through the plate and backing. The acceptance of the lot depends on both shot getting through without cracks or deformation. An alternative test specifies that the shot must go through, whole or broken, 1 caliber of hard-faced plate at about 1,900 feet per second.[25]

CAPS.

About 1877, at Shoeburyness, a Palliser shot was fired at a steel-faced plate with a $2\frac{1}{2}$-inch wrought-iron plate in front. It was found that the effect of the steel as regards the breaking

[25] "The Engineer." "American Armor-piercing Projectiles," November 19th, 1897.

up of the head of the projectile was completely neutralized, and its resistance consequently diminished to even less than that of ordinary soft-iron armor. English then suggested that a cap of wrought iron placed over the nose of the shot, so as to be cushioned between the shot and the plate on impact, would probably produce the same result; in accordance with this suggestion four capped 9-inch Palliser were fired in 1879. The cap used was similar to the present Russian type, and had a truncated nose; but the anticipated results were not realized. In the light of subsequent information it is not at all clear whether this failure was due to the insecure means of attachment, the low striking velocity—1,500 feet per second—or the already damaged plate against which the shots were tried. The cap fitted over the head of the projectile, and extended as far as the extractor holes, into which it was secured by screws. Above the point of the projectile the cap was $2\frac{1}{4}$ inches in thickness, and its point was truncated to a breadth of $2\frac{1}{2}$ inches. It was found that the capped shot were in no way superior to the ordinary pattern, and no further experiments were carried out.[26] In 1894, the Russian Admiral Makaroff was firing capped shot against harder plates than the old compound, and with recorded success. The caps were reputed to be of steel, and were termed "magnetic," probably because of a magnetic means of attachment. Since that time experiments have proceeded in many countries, but not always with consistent success. Capped shot have been known to fail against plates at striking velocities suitable for uncapped shot to succeed. In August, 1894, two 6-inch chrome-steel Firth capped projectiles perforated a 6-inch Harveyized plate at about 1,800 feet per second;[27] but in July of the same year a Hadfield uncapped shot pierced a similar plate at 1,825 feet per second;[28] so that in this country, up to 1894, and against Harveyized plates, the cap had not demonstrated its value. In 1900, however, Hadfield was able to produce cap-

[26] Cunhill. "Experiments with Armor-piercing Projectiles." "Proceedings, "R. A. I., Vol. XI, No. 7.
[27] "The Engineer," September 17th, 1897.
[28] Hadfield. "The Engineer," September 24th, 1897.

ped projectiles which, with a slightly higher velocity than the average usually employed, could readily perforate hard-faced Krupp plates.[29]

With steadier shot, improved manufacturing processes and materials, and at high striking velocities it has now come to be generally acknowledged that a cap adds from about $\frac{1}{4}$ to $\frac{1}{6}$ to the penetrative power against modern Krupp or Terni plates, at 1,800 feet per second. The cap, however, becomes rapidly less efficient as the angle of incidence is increased, so much so that beyond about 30 degrees the cap is useless. This fact needs to be insisted upon. It has been estimated that less than 10 per cent. of shots fired from a moving ship at sea will hit a target normally,[30] and considering normal impact with reference to the horizontal and vertical plane, it will easily be seen that at any range normal impact is altogether out of the question. With guns from high-sited batteries impact must always be very oblique.

There is a great variety of caps in use at the present time, and as great a variety in the means of their attachment to the projectile—Fig. 2. Hadfield's Heclon shell have the cap hammered into thumb-like recesses milled on the nose of the projectile. Johnson forces the cap into an annular groove with a few notches to prevent rotation. Firth sweats the cap to the nose with low-fusing-point solder, and also by running white metal into a groove in the cap and a corresponding groove on the nose. The Firth-Sterling caps are secured by small pieces of iron rod driven through holes in the cap into a groove on the nose; while many foreign examples are secured, like the experimental caps of English, by screws. Each of these means of attachment seems to fulfil the purpose for which it was designed, and whether the nose is grooved or not seems to be of little importance, since when shells break upon impact they do not generally do so through the groove as a weak section. The weak section seems to be determined by the initial stresses set up in the hardening process. In

[29] Hadfield. "Nature," January 25th, 1900.
[30] "Transactions" Institution of Naval Architects, Vol. XVIII, page 132.

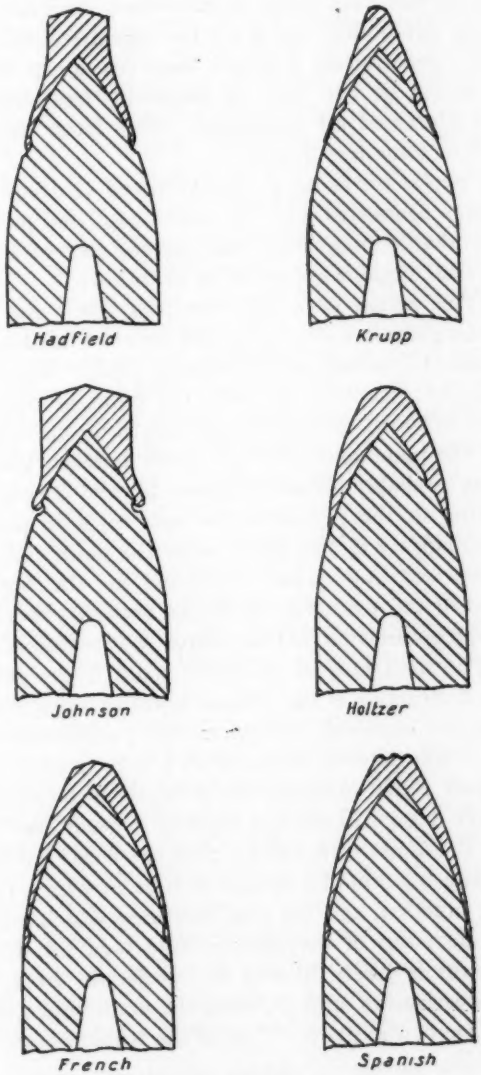


Fig. 2.—CAPPED SHOT.

many cases the caps seem to have been designed to secure something patentable, and if all the caps are equally efficient as regards penetration, it would seem that a cap such as the Krupp would be the best, as involving less shortening of range by increased air resistance. This latter aspect of the matter is of some importance.

Many explanations have been proposed to account for the phenomena connected with the use of caps, and putting aside the unthinkable suggestion that the cap acts as a lubricant, the principal notions appear to be as follows:

(1) "The action of a cap was that when the projectile struck the plate it dished it at the point of impact, and perhaps strained it nearly up to its elastic limit. The first thing to touch the plate was the cap; the point of the projectile had then to travel practically the length of the cap before it touched the plate, and when the point touched the plate the latter was probably already strained beyond its elastic limit. That being so, the projectile had easy work in piercing the very hard surface of the plate; once the point had got in it was supported; and it only remained for the high energy contained in the projectile to do the necessary work of completing the perforation, if that energy was sufficient.[31]"

(2) "The hard face of an armor plate is designed to initiate the destruction of the delicate point of a projectile before the latter has obtained any appreciable penetration at all, for directly it has entered, as much as $\frac{1}{8}$ inch even, it obtains a side support which increases the difficulty of breaking it, and the further it goes in the less support it needs, and the more it gets. It follows from this (1) that the hard face has only a very minute fraction of a second of time in which to perform its main function, and (2) that anything which will enable the extreme point of the shot to hold together during this brief instant is likely to save the projectile from the fatal initial pulverization, and to defeat the main object of hardening the face of the plate. This is the whole *raison d'être* of a cap.[32]"

[31] Dawson. "Armor-piercing Projectiles." "Proceedings" Inst. C. E., February 10th, 1903.
[32] Tresidder. "Brassey's Annual," 1905, page 347. Ewing. British Association, 1906. Discussion on "Armor Plates," by Edwards.

(3) "The causes of the rupture of the ogival head, and the means of preventing it, require delicate examination. Let us consider the case of the rupture of a projectile of good quality. . . . The point of the head was flattened by the blow, and took the form of a punch, which latter commenced the penetration; the ogival head afterwards entered the metal like a wedge. In proportion as the ogival head penetrated, the portion enclosed in the plate was violently compressed, whilst the part which remained outside underwent no pressure. This sudden passage from one state to the other, following the line of contact with the exterior surface of the plate, creates a critical state for such a rigid metal as chrome-steel; if this separation propagates itself into the interior rupture will take place. A remedy has been sought in the use of a more or less flexible and ductile envelope, which, placed upon the projectile, would transmit the pressure to it, and at the same time spread it out in a slightly graduated manner. This covering, or cap, of soft steel thus fulfils, from the point of view of the distribution of the pressures, an analogous part to that which is played, from the point of view of the distribution of the metal, by the curve which unites the two portions, of different diameters, of an axle, for the purpose of preventing the formation of a crack in the angle which the curves do away with.[33]"

Regarding these explanations it may be noticed that the first assumes that a soft-iron projectile fired against a face-hardened plate can, in the earliest stages of impact, strain the face of the plate up to its elastic limit, so that presumably in the immediate neighborhood of the point of impact the face is in a plastic or semi-plastic state. Now for the lowest striking velocity at which a cap is known to be effective, this assumption would not be generally admitted. And, further, if the cap strains the face in this manner the necessity for a very hard nose to the projectile is not at all clear; it seems reasonable to suppose that an ordinary steel nose would be

[33] Bertin. "Hardened Plates and Broken Projectiles." "Transactions" Institution of Naval Architects, July 6th, 1897.

sufficient to fracture already plastic material. Moreover, phenomena almost identical with that associated with caps can be produced by placing a layer of soft iron, or even copper or lead, in front of the plate instead of on the projectile; and evidently in this case the action can hardly be one of producing initial plasticity of the hard face.

The second explanation is typical of the one current in text-books. As far as I can understand its terms, it refers to the great resistance offered by materials to volumetric strain. That portion of the nose going through the cap is under lateral pressure arising from the resistance offered by the cap to the wedge action of the projectile. It is also under end pressure, at the extreme point, the measure of the initial resistance of the plate, and at the other end the pressure corresponding to the force of retardation acting on the remainder of the projectile. The front few inches of the nose is thus under a sort of fluid stress, and the only probable strain is that due to a change of volume—in other words, a crushing strain against which the material can offer very great resistance. The nose thus retains its shape, and is able to produce the intense local strain on the face which is favorable to fracture of the plate. But this explanation fails to account for the general experience that caps add nothing to the penetrative power below a striking velocity of about 1,700 feet to 1,800 feet per second, although this velocity is above that necessary to pulverize the bare point of the projectile. It also throws no light on the behavior of small capped projectiles, which, generally speaking, seem to gain little by the use of a cap, or on the behavior of capped shell against plates differing considerably from one caliber in thickness. M. Bertin's explanation obviously refers to the fracture of the head of the projectile rather than to the pulverization of its point; and the question at issue is not a consideration of the cause of fracture after the point has entered the plate, but how the point is able to enter at all and preserve its original shape.

The principal objection to these explanations appears to be

that they attempt to account for the phenomena resulting from a very complex play of forces, as if one out of many considerations affected the whole result. It has to be remembered that in the head of an armor-piercing projectile, where in a small mass we have a large variation of cross-section, initial stresses of considerable magnitude exist; and anything which can reduce the time rate at which the velocity of the projectile is reduced in the stage immediate on impact, is favorable to the instability of the material of the nose remaining undisturbed. The cap would, from this point of view, act as a cushion, reducing the maximum value of the retarding resistance set up by the face of the plate. In the next place, the cap acts in a manner favorable to the nose-resisting fracture, by transmitting the strain energy from the small mass at the point to the larger mass of the body of the projectile. As far as I understand Bertin, this is the main feature of the action of a cap. Now, it is known that at the short ranges usually adopted at proof butts, projectiles are not generally moving in a steady condition; that it to say, they have angular motion about an axis inclined to the axis of figure. As a consequence, armor-piercing projectiles at proof nearly always strike the plate with an incidence of a few degrees, and this oblique action, being accompanied by cross strain on the nose, is most favorable to the initial fracture of an unprotected point. Whether the pulverizing of the broken point is produced directly on impact, or whether it results subsequently from the crushing action of the remainder of the projectile, is not known. The cap, in being the means of transmitting this lateral strain from the point to the body of the head, favors the head remaining intact. Steel caps would obviously be better than soft-iron caps if their action has anything to do with the transmission of strain energy from the point to the body of the head.

The experience at our disposal lends support to the idea that the problem of armor-piercing projectiles is the problem of finding a suitable shape to the nose.

IMPACT.

The general problem of impact is one of great complexity, particularly so when the immediate aspect is the strain effect on the colliding bodies, and not the translational and rotational motions which the collision may ultimately produce. Some quantitative statements founded on more or less exact elastic theory have recently been furnished,[34] but they throw little light on the armor problem, because here we are concerned with a final state of fracture and not with a final condition of strain. And if it were otherwise, we should still have considerable difficulty in the approximate application of the ordinary elastic theory to a modern armor plate, which, being by no means isotropic, presents a marked variation in physical properties throughout its thickness. It is hence not a matter of surprise that the theory of armor under attack is confined to quantitative statements derived from empirical rules, which, with one unjustifiable exception, are candidly without theoretical basis.[35] These rules are confined to the solution of two classes of practical problems—the determination of the thickness of a plate just perforatable by a certain projectile, striking normally at a certain velocity; and the comparison of the merits of two plates, according to their behavior against the same nature of projectile. The first class is solved by reference to one of the many rules that a maker finds best conforms to his own practice, and the second by the introduction of a ballistic factor of merit, used in conjunction with a formula, such as that of Tresidder, believed to express the behavior of homogeneous unbacked wrought-iron plates; or a formula such as that of Jacob de Marre, for the homogeneous steel plates made by Barba at Le Creusot. When we remember the complex and difficult treatment of large armor plates, and the secrecy preserved with regard to the processes, and composition of materials, &c., it will be apparent that considerable diversity is to be expected in the published results regarding armor and projectiles. If in

[34] Smith. "Impact, Direct and Oblique." "The Engineer," March 10th, 1906.
[35] Tresidder. "The Engineer," July 24th, 1896.

nothing else, this diversity is most clearly shown in the great variety of empirical rules that are in use. It appears to be generally assumed that the thickness of a hard-faced plate just perforated, under normal impact, can be expressed as a single term involving functions of the weight, striking velocity and diameter of the projectile; and the changes are rung in every conceivable way, although it is well known that considerations other than these are effective in deciding whether the plate is to be defeated.

Krupp makes . . . . . $t^2 \propto \frac{w\,v^2}{d}$;

Davis makes . . . . . $t^{1.5} \propto \frac{w}{d} v^2$;

Tresidder makes . . . . . $t^2 \propto \frac{w}{d} v^3$;

Vickers (Russian) makes . . . . . $t^{.7} \propto \frac{w^{.5}}{d^{.75}} v$;

t, being the thickness of hard-faced plate just perforated by a projectile of weight w, diameter d, and with striking velocity v. In fact, given a table representing fair average values of t and v as obtained from actual practice, then one has only to assume the form of any of the variables t, w, v and d, to make a very large number of these rules accurate to about 5 per cent., which is well within the limit of experimental error. I have plotted the above formula and taken a mean curve; assuming w and d to be connected by the relation already given, I find for service uncapped armor-piercing projectiles—

$$t = \frac{d^{\frac{1}{2}}\, v^{\frac{1}{2}}}{12{,}000},$$

and for capped projectiles—

$$t = \frac{d^{\frac{1}{2}}\, v^{\frac{1}{2}}}{10{,}000}.$$

Major Edwards, R. A., on the assumption that

$$\frac{w}{d^3} = .46$$

has pointed out that the formula for Krupp steel can be reduced to the form—

$$t=\frac{D\,v}{2{,}200},$$

for uncapped projectiles, and that a rough rule for capped projectiles is

$$t=\frac{v\,d}{1{,}950}.$$

There are several considerations which limit the utility of these rules. In the first place, they are only applicable to complete perforation by unbroken projectiles that have inappreciable remaining velocity after passing through the plate, and as most of the information on which the above formulæ have been built up has been obtained from tests in which the diameter of the shot was not widely different from the thickness of the plate, these rules should only be considered to have application to those cases in which the ratio $\frac{t}{d}$ approximates to unity. Against hard-faced plates the rules cannot be used for partial penetration. Thus, a 9.2 uncapped shot which will perforate an 8-inch Krupp plate at about 1,900 feet per second must not be expected to enter 8-inch in a 12-inch plate at the same velocity. Considerations of stiffness and thickness of cemented face affect the result, as will be indicated later. And again, in the trials, to cover the results of which the above rules were devised, several factors of importance were generally unknown. The striking velocity was usually calculated from the charge, and not experimentally observed. No cards were taken near the plate to ascertain whether, at the short proof ranges the initial unsteadiness of the shot had been damped by air friction, in spite of the fact that shots frequently hit the plate at points considerably out of the line of sight.[36]

In cases where the plate was perforated the effect of the backing, or what the remaining velocity would have been

[36] Cranz and Koch. "Untersuchungen über die Vibration des Gewehrlaufs." (München, 1899.)

without backing, was not determined; and in cases where complete perforation of the plate did not take place, what increase in striking velocity would have brought about perforation; although such experience with hard-faced plates as we possess goes to show that almost up to the point of yielding the plates show little or no indication of the approach to yielding.[37] Bertin, in his elegant discussion, assumed the rough rule, that the resistance of the backing was equivalent to an increased thickness of the plate equal to one-tenth the thickness of the backing.[38] The spring of the target and the permanent distortion of the plate were usually not recorded, or simply mentioned as an incident of the test. In addition, we must allow that plates from the same maker would show at least 5 per cent. variation under the ordinary tensile and impact tests, and hence we ought not to expect any of the above rules to show a greater consistency than such as lies within a 10 per cent. limit of variation. The behavior of modern armor is hence not rationally expressed by the usual tables, in which thicknesses of plate, just perforable, are given to two significant figures.

The most important practical problem connected with the attack of armor is the comparison of the qualities of defence of different plates. Two methods of meeting this problem are in common use. The first proceeds by a determination of the ratio between the thickness of a standard wrought-iron or steel plate, and the thickness of the hard-faced plate, that is just perforated by a certain projectile with a certain velocity. The second, by a determination of the ratio between the striking velocities necessary with the same projectile, in order to perforate the same thickness of the different plates. The comparison of thicknesses is the method adopted in this country, the standard plate being unbacked wrought iron, as represented by Tresidder's formula,

$$t^2 = \frac{w v^3}{d} \times \frac{1}{\log 8.8410}.$$

[37] White. "Transactions" Institution of Naval Architects, July 6th, 1897.
[38] Bertin. "Transactions" Institution of Naval Architects, July 6th, 1897.

On the Continent the comparison of striking velocities finds favor, the standard plate being soft iron, as represented by the Gâvre's formula, or more generally Le Creusot steel, as represented by the de Marre formula, which in French units may be written:

$$t^{1.4} = \left(\frac{1}{1{,}530^2} \frac{wv^2}{d^{1.5}}\right).$$

Neither of these methods can be considered as offering a scientific treatment of the subject, although they have come to have extensive practical adoption. Nevertheless, they make direct appeal to the naval architect and to the gunner. A comparison of defensive qualities measured in terms of thickness, and hence, for a given area of protection, in terms of weight, furnishes an immediate account of the minimum weight of armor to be carried to protect flotation, or for any other purpose, such as protecting gun bases, funnel bases, uptakes, &c., that may be thought desirable; whereas a comparison of these qualities in terms of striking-velocities furnishes an account as to ranges beyond which certain types or thicknesses of armor cannot be defeated by a specified nature of attack.

To facilitate the comparison of plates according to the English system, Tresidder's formula for unbacked wrought iron is generally taken as a basis, and used in conjunction with a ballistic term called the "factor of merit." This factor is defined as the ratio, R, between the thickness of plates, one of wrought iron and the other of hardened steel, both of which are just perforated by a given projectile with a given striking velocity. Another term, called the "factor of penetration," is due to Wolley-Dod. It defines the ratio between the thickness of wrought iron just perforated by a given round and the thickness of the plate against which the round is fired.

It is not at all clear what sort of comparative value we are to consider this factor of merit to possess; whether, in fact, we are to consider the quality of a plate as entirely furnished

by its factor of merit. Thus, supposing as a result of improved processes the factor of merit of a certain thickness of plate against a certain projectile went from 2 to 3, are we to consider that the improved plate offers half as much resistance again (resistance being understood to mean work spent in penetration) as the original plate? If so, a comparison of the plates by the continental method of velocity ratios should presumably show a like increase. But practical experience shows that there is no simple relation between the increase in quality, as measured by the ratio of thicknesses, and that measured by the ratio of velocities. Furthermore, the figure of merit is modified by the ratio between the thickness of the plate and the diameter of the projectile, in quite a different way from that of the de Marre coefficient. Hamilton has recently endeavored to insert considerations of this ratio in the ordinary Tresidder formula, but the data on which his formula are based are very restricted. He deduces,[39]

$$F = 2\left\{1 + .0014\left(13 - d\right)^2\left(\frac{t}{d} - 1\right)\right\},$$

F being the figure of merit, t the thickness of the plate, and d the diameter of the projectile. Introduced into Tresidder's formula for wrought iron, Hamilton gives the following formula for K.C. plates by capped A.P. projectiles, under normal impact:

$$\left(\frac{t}{d}\right)^2 F^2 = \frac{W}{d^3} v^3 \log^{-1}\left(0.55694 - 10\right).$$

The general phenomena arising from the impact of a shot on an armor plate appear to be as follows: As soon as the projectile touches the plate it suddenly experiences enormously increased resistance, and its rate of retardation is suddenly increased. The measure of the retarding force, in a direction parallel to that of motion, is directly proportional to the time rate of destruction of momentum of the projectile at the instant considered. This force acting on the nose of the projectile, in strictly normal impact, results in a wave of

[39] Hamilton. "Perforation Formulas." "Journal" United States Artillery, Vol. XXV, No. 3.

compression traveling from the point of the projectile towards its base, the velocity of this wave being directly proportional to the modulus of elasticity, and inversely proportional to the density. At the instant when this wave first travels towards the base, the rear portion of the projectile is moving towards the plate at the same velocity that it had immediately before impact, so that shortly after impact the nose and the base are moving in opposite directions, and if the thrust corresponding to this motion cannot be carried by the walls of the shell the walls crack or set up.

The pressure between the projectile and the plate results in a similar state of compression spreading throughout the plate, and this wavelike mobility of the particles of a plate under severe impact is visible in many plates.[40] Some very perfect specimens of these curves have also been obtained from firings at official proof butts. The velocity with which these waves are propagated also depends on the elasticity and density of the plate.

The immediate effect of such a rapidly-applied force is, as we have seen, both on the projectile and plate, to cause intense local strain. The magnitude of the strain energy per unit volume can be calculated when we know the stress and strain throughout a small elementary mass. If f is the average compressive stress, and E the modulus of the material for shock loads, then the strain energy per unit volume is measured by $\frac{f^2}{2E}$. If the materials at any time become stressed to the point at which permanent set takes place, $\frac{f^2}{2E}$ measures what is called the "resilience." We thus see that the amount of strain energy per cubic inch that the projectile or the plate can possess without permanent deformation or fracture is perfectly definite, and would be ascertainable if the physical constants under the actual conditions of the applied loads could be determined. If, then, at any time stress accumulates so rapidly that the strain energy exceeds the resilience of the

[40] Hartmann. "Comptes-rendus de l'Académie des Sciences de Paris," March, 1894. Institute of Naval Architects, July 6th, 1897.

materials, semi-plastic yielding or fracture must ensue. On the other hand, if by employing suitable materials, or by their suitable arrangement, we can ensure the strain energy being dissipated so fast from the point of impact that its value at any place is less than the resilience, local set or fracture will have been prevented. It follows as an immediate consequence that the fracture of a body under a blow depends on the ability of the body to dissipate strain in such a way that the strain energy is always less than the resilience.

We have seen that strain energy is measured by the product of stress and strain, and since the proportionality of stress and strain continues much beyond the ordinary static elastic limit,[41] we may say that strain energy, as far as we are here concerned, is measured by the square of either stress or strain, and the possibility of fracture is represented by the square root of the strain energy per unit volume. If the material is brittle, there is a sort of molecular instability which causes fracture at one place to extend to all neighboring places. Such an effect was produced in the old compound plates, the steel face of which frequently cracked into many pieces, like a piece of glass. The same feature is often seen in modern Krupp plates where the cementation has been inefficiently performed, the face breaking up with the well-known conchoidal fracture characteristic of obsidian. In these cases a moderate supply of energy, delivered with great rapidity, is sufficient to produce extensive fracture; but as the material gets less and less brittle we require a larger amount of energy delivered to get fracture throughout the same volume. It is for this reason that in cutting wood we use a wooden mallet and a long wooden-headed chisel. The mallet and chisel here act as a reservoir of energy, and the energy of the blow is slowly delivered by the edge of the chisel to the wood, but the rate is sufficient to cause fracture immediately in front of that edge. If the wood became more rigid without increasing in strength, so that the energy delivered by the blow would

[41] Hopkinson. "Effects of Momentary Stresses in Metals." "Proceedings" Royal Society, February 16th, 1905.

be transmitted more rapidly to other parts of the material, then we should have to use a more rigid mallet and a more rigid chisel. In dealing with a metal such as steel, we have to ensure a still greater rate at which energy can be delivered by using a short rigid steel chisel and a steel hammer.

It will thus be seen that if the wave of compression passing along the projectile and through the plate gives rise to such an accumulation of strain that the strain energy is greater than the resilience, fracture of the projectile and penetration of the plate must take place. We thus see the immediate function of a hard face to the plate and a hard nose to the projectile, hardness being used to signify, not mere capability of being scratched by other materials, but a high modulus of elasticity. If the point of the projectile is of highly elastic material, such as tempered steel, the strain energy of compression is rapidly propagated to the body of the projectile, and the yielding of the point is prevented. Precisely similar conditions obtain with regard to the plate. A highly elastic face enables strain energy to be rapidly conveyed from the immediate locality of impact to other parts of the body of the plate, and the strain energy not being allowed to accumulate to the limit of resilience, fracture is prevented. Since the velocity with which waves travel from the point of impact is determined by the physical properties of the material of the plate, it follows that any projectile, however soft, can fracture a plate, however hard, if only it has sufficient striking velocity; for the force of impact can be increased indefinitely with the velocity, whereas the rate of propagation of strain is limited by the materials used. But with actual projectiles and actual plates any increase in striking velocity is in favor of the projectile breaking on impact, and a broken projectile is all in favor of the plate not being defeated.

Again, in general, the rapidity with which a body can propagate strain depends on the direction of the blow with regard to the configuration of the body. Thus, a steel rod can be much more easily broken by a transverse blow than by a longitudinal blow, because the rate at which strain energy

is conveyed from the point of impact to other parts of the rod is less when the direction of the blow is at right angles to the axis of the rod than when it is parallel to that axis. This fact explains the rapid diminution in the power of a projectile to perforate a plate at oblique impact. The blow on the nose can be imagined resolved into an axial blow and a transverse blow. The strain energy due to the transverse component is not propagated as fast as the strain energy due to the axial component, and fracture of the nose is favored. The converse holds with the plate. Striking a plate at an angle is equivalent to sending component compression waves at right angles to its face, and along its face; and these latter waves, traveling at a greater rate than the former, tend to distribute the strain energy throughout a large volume, and thus prevent fracture, as the plate behaves like a rod, being more easily broken by a blow at right angles to its length than in the direction of its length.

It would thus appear that the usual methods of dealing with oblique impact are not rational; the problem is much more complex than the usual assumptions would lead us to believe. Treating the obliquely presented plate as one normally presented, but thicker in the inverse ratio to the cosine of the angle of obliquity, is a method for which we can find no theoretical sanction. Tresidder has proposed the formula,[42]

$$v = \cos\frac{\delta}{2}3\sqrt{\frac{C^2t^2\,D \times \log -^1 8.8410}{W\cos^2\theta}},$$

which we may take to be an empirical rule awaiting trial, according to which the figure of merit of the oblique thickness is taken, and the mean direction of the hole in the plate to be halfway between the direction of impact and the normal to the plate. According to some trials of 6-inch capped A.P. projectiles by the Bethlehem Steel Company, quoted by Hamilton, the return to the normal was greatly increased by the use of a cap, and he proposes the following formula, in

[42] Tresidder. "Brassey's Annual," 1905, page 345.

which the return to the normal is taken as two-thirds the angle of obliquity:

$$\log P_\theta + \log d^3 + 9.443 = \log W + \log V_\theta^3,$$

in which P_θ is found with $\left(\frac{t}{d}\sec\theta\right)$ as argument, and

$$V_\theta = V\sec\frac{2}{3}\theta.$$

The method by which only the normal component of the striking velocity is considered as effective in producing penetration is obviously incorrect, since the transverse component has a most important effect in putting cross strain on the nose of the projectile, a condition most favorable for the fracture of the nose, as well as setting the face of the plate, at the point of impact, into a vibratory state, and thus relieving the face at the point of impact.

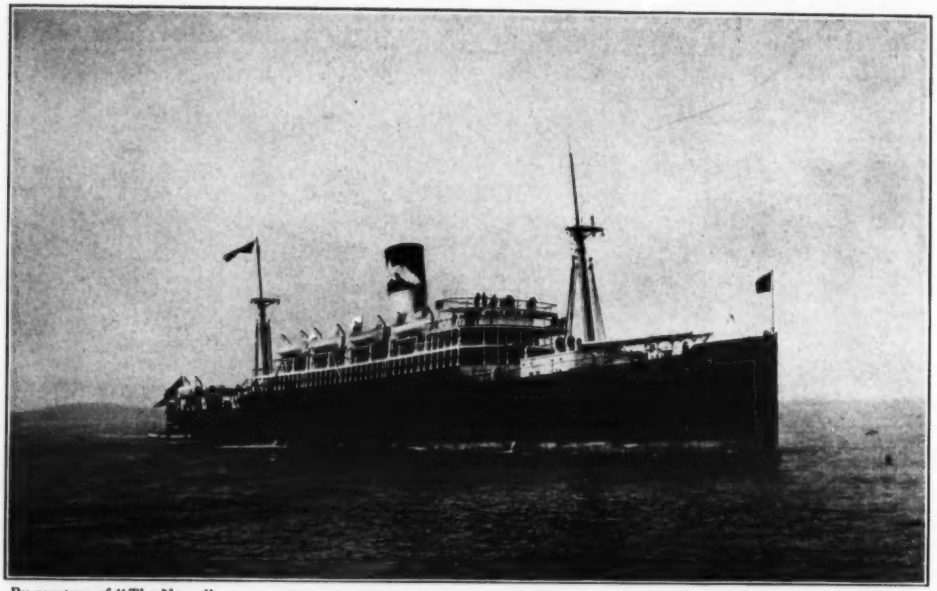
With regard to the action of a projectile, a somewhat artificial distinction has been made between a blunt and a pointed projectile. Thus Orde-Browne has it—"A plate may be perforated by 'boring' and 'punching.' An armor piercer, unless prevented, drives its sharp point through a plate at the point of impact, opening the metal outwards, and so clearing a path for the shot's body to follow. This process may be called 'boring' a hole. A blunt projectile cannot do this. It can only perforate by driving a disc of plate out in front of it. This is the way that a common shell with a plain head and a nose fuse perforates, and it may be termed 'punching,' and is resisted by the plate's tenacity and toughness.[43]" To an engineer this distinction must appear somewhat unreal. We invariably associate boring action with rotation, and in this respect projectiles, whatever be the shape of the nose, are similar. Even if the terms are considered to make reference to the character of the holes, they still seem to be inapplicable, since a punch will make a clean round hole in a plate if the latter is not brittle. The action appears to be one of degree,

[43] Orde-Browne. "Brassey's Annual." "Armor." 1899.

and not one of kind, as the usual reference implies. A pointed projectile is favorable to an initial localized strain, and hence to early fracture of the skin of the plate, which, owing to the molecular instability before mentioned, allows the fracture rapidly to extend. A flat-headed projectile acting in the early stages over a comparatively large area meets with a greater total resistance, but the resistance is less intense, and the corresponding strain is also less intense. Considering the time average value of the resistance, we see that in the case of the pointed projectile the portion of the plate just touching the nose is fractured in the very earliest stages of impact, before the waves of compression have had time to travel very far from that point, whereas with a flat-headed projectile fracture of the plate occurs at a later period, and the waves of compression have had time to travel from the area of impact to other portions of the plate, which are also fractured if the strain energy per cubic inch produced by the compression wave is equal to the resilience of the material.

The appearance of the hole made in the various plates presents some general characteristics. With the old iron or soft steel plates, a large bulge was first made at the back, followed by a tear extending from about the center of the bulge by three or four cracks to the edge of the hole, the edges being usually turned back. With hard, homogeneous steel plates the effect was quite different, and was much the same as that experienced with Gruson chilled cast-iron cupolas. The fracture seemed in no way to depend on the diameter of the shot, due, no doubt, to the reason already given—the rapid spread of the fracture with brittle metals. In the case of a modern Krupp or Terni plate quite another effect is met with. Elastic deformations of the plate may obtain a considerable value, and frequently the diameter of the hole is less than the diameter of the projectile. A large, slightly conical disc is punched out, such as would be produced by a considerable elastic deformation, accompanied by a tearing of the hole at its edges. Capped projectiles usually produce holes slightly larger than the projectile, and without surface scaling. At

oblique impact, if the plate is one just matched by the projectile at normal impact, scoops a few inches in depth are usually made, but if the plate is overmatched the capped projectile makes its characteristic hole, the general direction of which is inclined to the plate at a smaller angle than the angle of incidence.



By courtesy of "The Navy."

SOUTHERN PACIFIC S. S. "CREOLE," FITTED WITH CURTIS ENGINES.

BUILDERS' TRIALS OF CURTIS TURBINE STEAMER *CREOLE*.

By Chas. B. Edwards, Associate.

The *Creole* is a first-class combined passenger and freight steamer built by the Fore River Shipbuilding Company for the Southern Pacific R. R. Co. She is fitted with twin-screw Curtis turbines, and the builders' guarantee calls, for a speed of 16 knots at 10,000 tons displacement. She can carry 420 passengers of all classes and 3,000 tons of freight. Figure 1 is a photograph of the vessel.

HULL.

The following are the principal dimensions of the vessel:

Length over all, feet	440
Length between perpendiculars, feet	416
Breadth, molded, feet	53
Depth, molded to upper deck, feet	37
Draught at 10,000 tons displacement, feet and inches	24–10
Coefficient of fineness, block	.645

BOILERS.

The vessel is fitted with ten Babcock & Wilcox water-tube boilers with superheaters, using natural draft. There is one smoke pipe 94 feet 4 inches in height and 153 square feet area. The total capacity of the ten boilers is:

Grate surface, square feet	770
Heating surface, square feet	28,500
Superheating surface, square feet	4,350
Ratio grate to heating	37
Working pressure, pounds	250

TURBINES.

The vessel is fitted with two 120-inch diameter, seven-stage Curtis marine reversible turbines, rated at 4,000 brake horsepower each, or 8,000 H.P. aggregate. Figure 2 is a view of the engine room looking aft, and Figure 3 shows one of the turbines with the wheels ready to assemble in the shop.

PRINCIPLE OF OPERATION.

The Curtis turbine is of the impulse type, and, in order to reduce the economical speed of rotation to a point suitable for direct driving of marine propellers, the turbine is divided into several pressure stages, and each pressure stage is provided with several rows of revolving buckets. The simplest possible impulse turbine would consist of a single wheel having a single row of buckets on which a jet of steam is directed by a suitable nozzle. Examples of such a turbine are the DeLaval steam turbine and the Pelton water wheel.

A simple single-wheel impulse turbine has its maximum economy when the buckets move with a velocity of approximately half that of the jet of steam. A jet of high-pressure steam, in a proper nozzle, will attain a velocity of about 4,000 feet per second when discharging into a vacuum, which would require the enormous speed of 2,000 feet per second for the buckets to obtain the maximum economy. This speed would require such a high number of revolutions that it would render direct connection to a propeller impossible, and would also produce centrifugal forces too great to be properly handled.

By using several wheels on the same shaft, each one utilizing the exhaust from the one preceding, the pressure drop and corresponding steam-jet velocity to be handled by each wheel is greatly lessened. Thus by using eight wheels, each in a separate chamber with its own nozzle, and so proportioning the steam pressure that the available energy is equally divided, the steam-jet velocity is reduced to about 1,400 feet per second and the bucket speed to 700 feet per second. This,

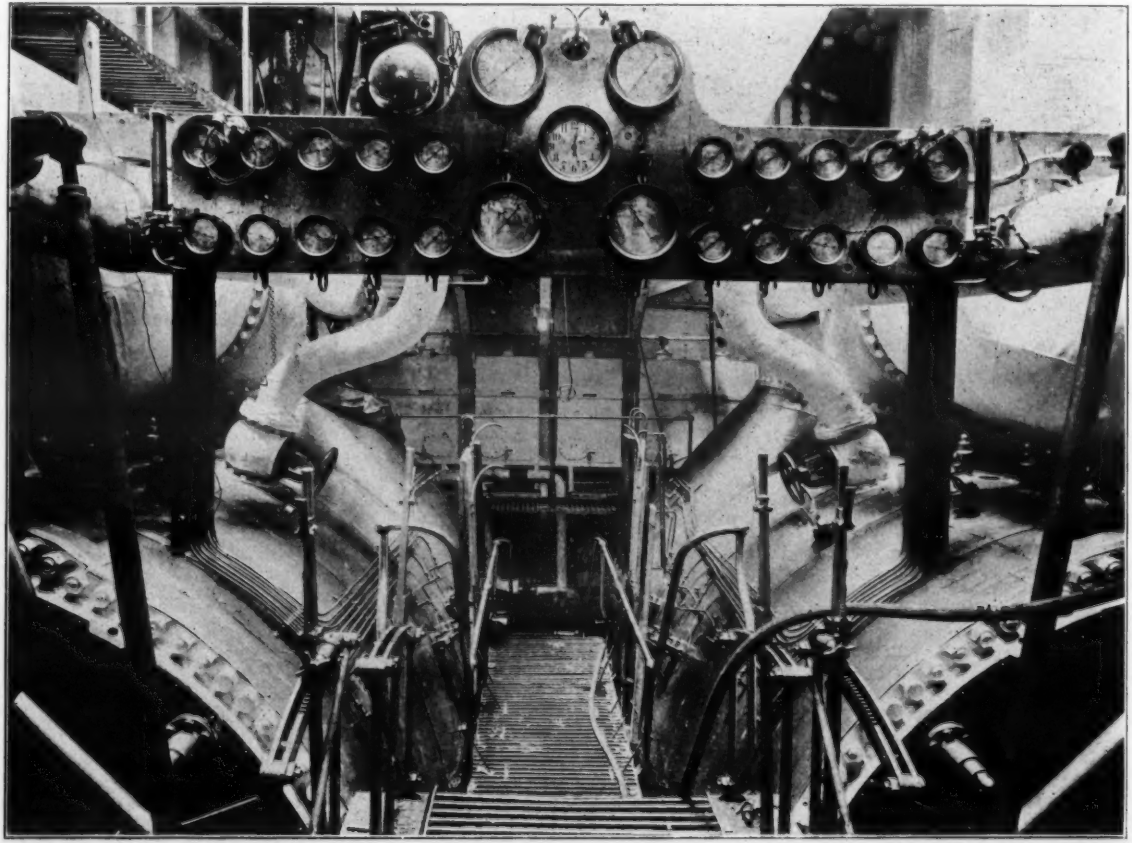


Fig. 2.—View in Engine Room, Looking Aft.

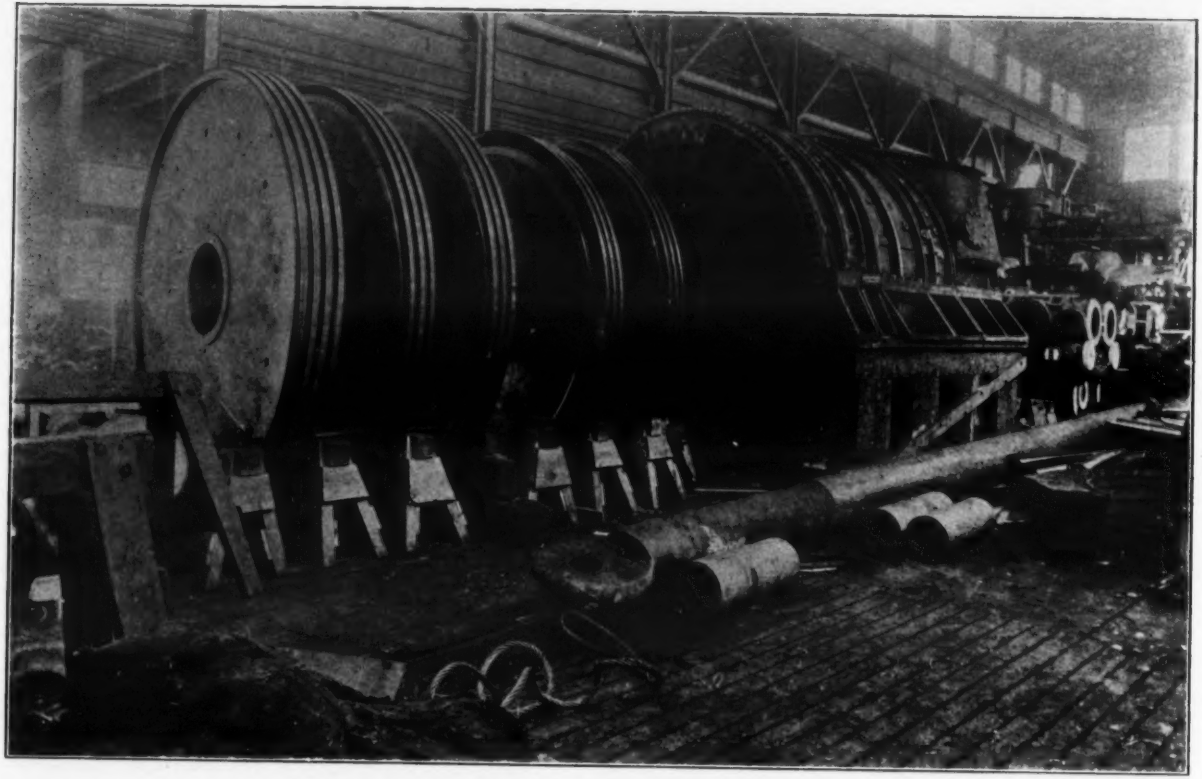


Fig. 3.—Turbine Casing with Wheels Ready to Assemble.

however, is still too high for satisfactory marine work, and, in order to lower it, each wheel is provided with three rows of moving buckets instead of only one. Each row of moving buckets will take out from the jet velocity approximately twice its own speed, so that the use of three rows on each of eight wheels will give a bucket speed of 230 feet per second. This speed can be used satisfactorily on high-speed vessels, but as the economy curve is quite flat at the point of maximum efficiency, the speed can be somewhat reduced without greatly reducing the economy, giving a fair working speed of about 160 feet per second. The above described method of reducing the economical bucket speed by means of several separate stages, each having several rows of buckets, is illustrated in diagrammatic form in Figure 4, which shows the first two

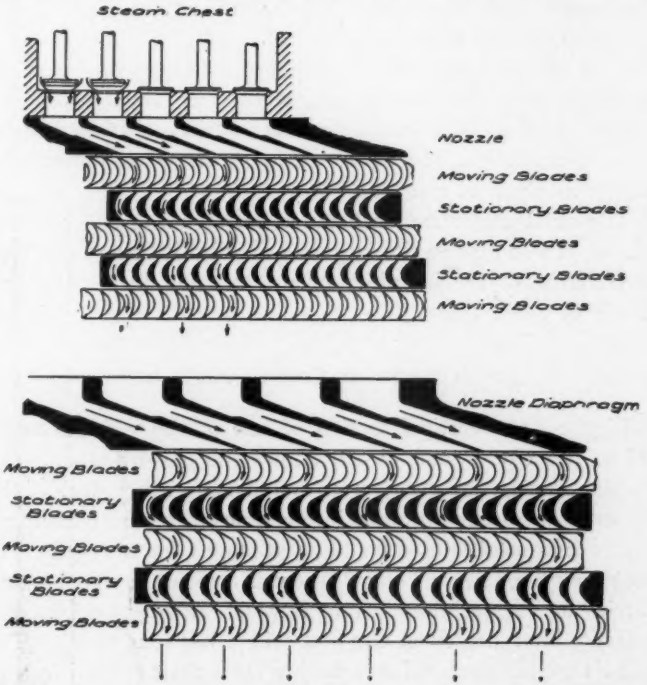


Fig. 4.—Diagram of Nozzles and Buckets in Ctrtis Steam Turbine.

stages of a turbine of which the remaining stages would be similar. It is seen that between each row of moving blades there is a row of stationary blades, the object of which is to reverse the direction of flow of the steam as it leaves one row of moving blades and direct it into the next row of moving blades. The arrows show the direction of the steam as it passes through the nozzle and then through the moving and stationary blades.

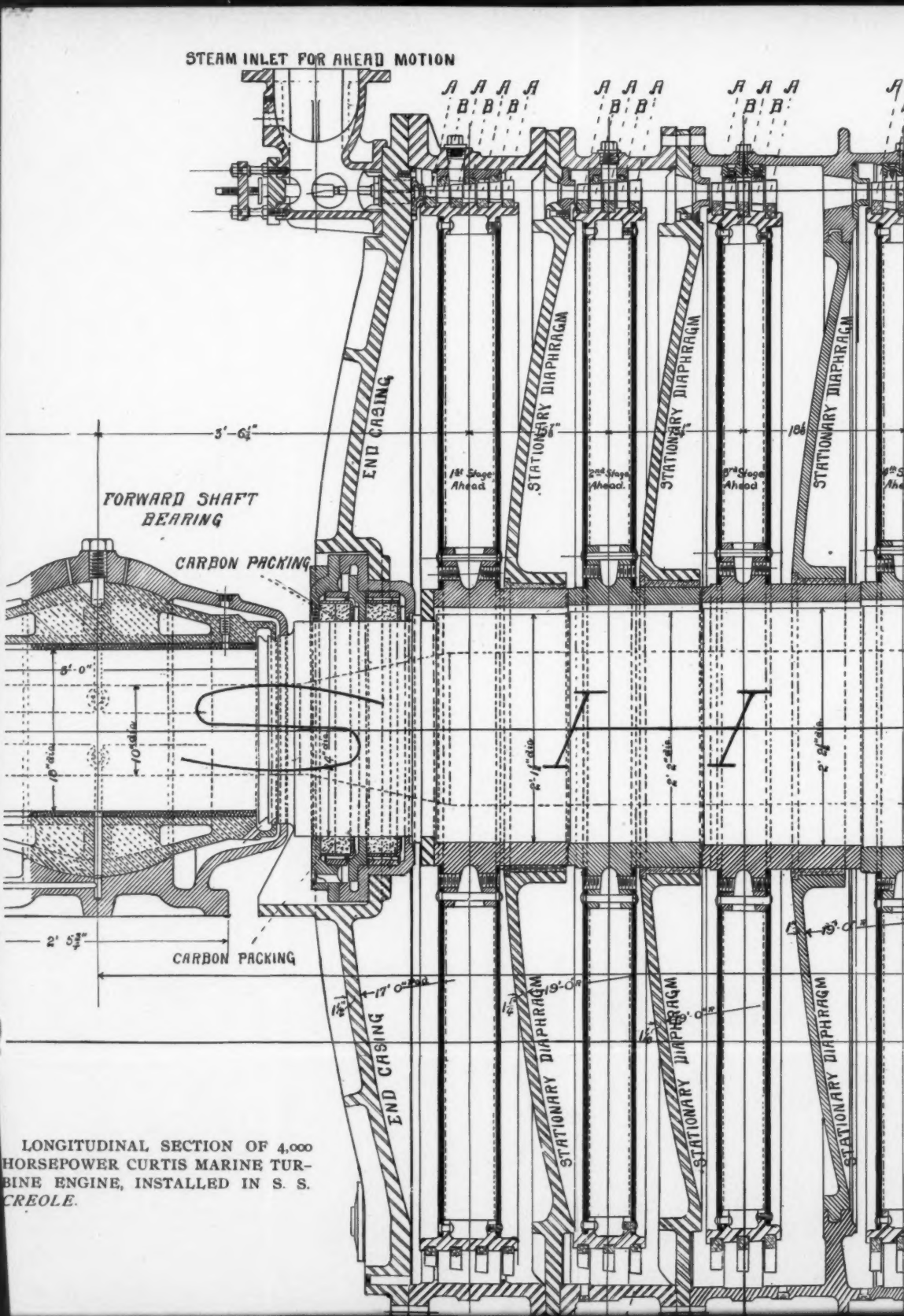
When the steam passes through a nozzle its pressure drops and it gains velocity; this velocity is absorbed by the moving buckets, thus converting energy of the steam into mechanical work. As the steam passes from stage to stage it drops in pressure, and when it passes from one row of buckets to the next in the same stage it loses velocity, but the pressure is the same in any one stage in all the bucket rows, and is also the same on both sides of the wheel.

This uniform pressure in all parts of stage obviates the necessity of small radial clearance between the ends of the blades and the casing, since there is no leakage around them. This clearance is, therefore, made as large as desired for proper mechanical construction.

The axial clearance, or distance between the edges of the blades, is also made quite large, its only limit being the slight disturbance of the jet which occurs at some distance from a nozzle or guide blade.

DESCRIPTION OF CURTIS TURBINES.

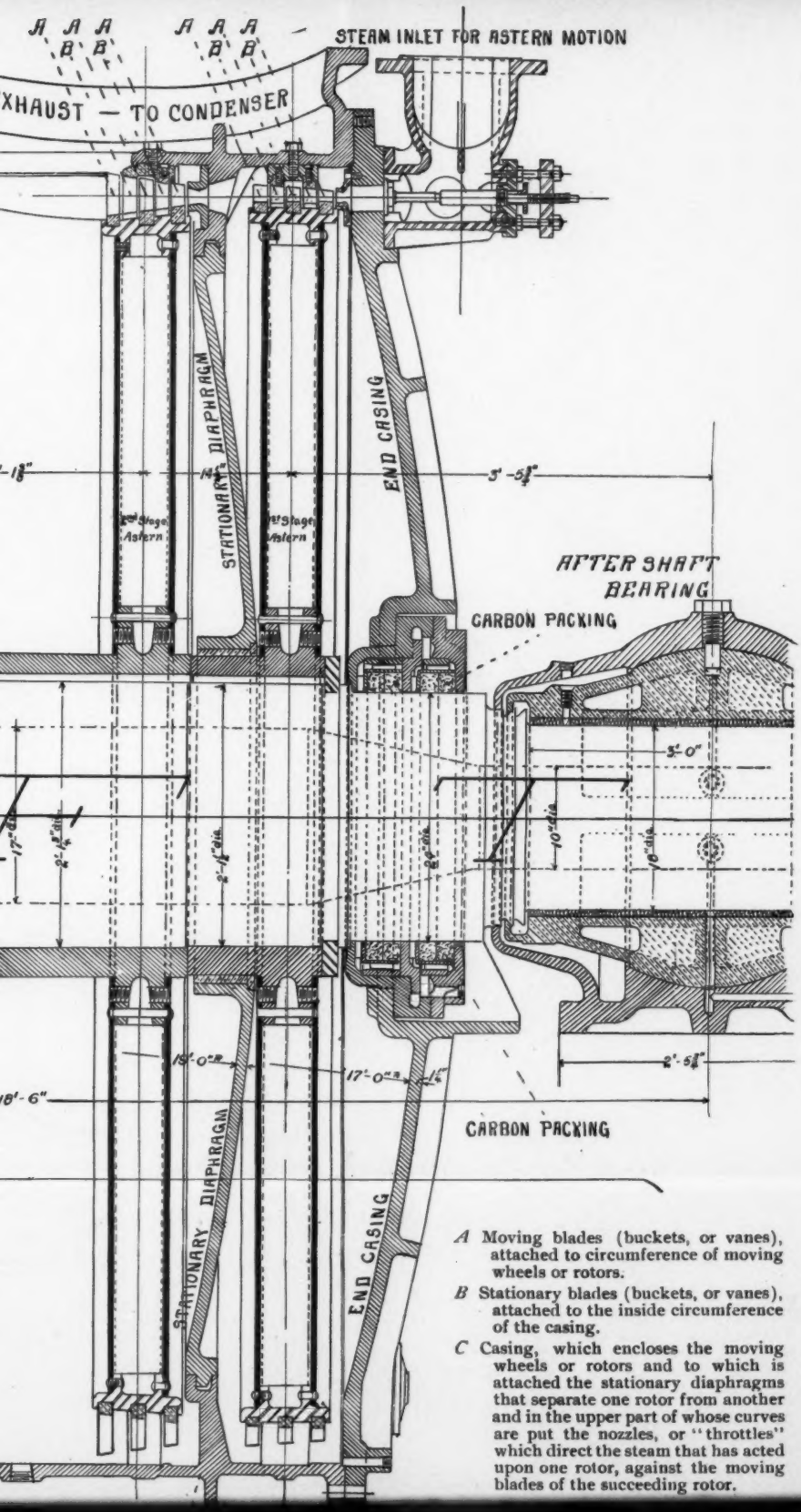
Figure 5 is a vertical section of one of the turbines as fitted to the *Creole*, and clearly shows the construction. The turbine consists of a cast-iron cylindrical casing divided by dished cast-iron diaphragms into a series of separate compartments (cast steel is used for two first stages, for strength). In each compartment or "stage" there is a separate wheel which carries on its periphery three rows of moving buckets (for reasons later described the first wheel has four rows). The wheels are all mounted on a hollow steel shaft carried by self-



LONGITUDINAL SECTION OF 4,000 HORSEPOWER CURTIS MARINE TURBINE ENGINE, INSTALLED IN S. S. *CREOLE*.

EXHAUST — TO COND
STATIONARY DIAPHRAGM
4th Stage Ahead.
5th Stage Ahead.
6th Stage Ahead.
7th Stage Ahead.
1st Stage Astern.
19"
2'-13"
2'-4" dia.
2'-3" dia.
9'-10" dia.
6'-2½" dia.
9'-6½"
17" dia.
2'-1" dia.
13'-0"
19'-0"
22'-0"
18'-6"

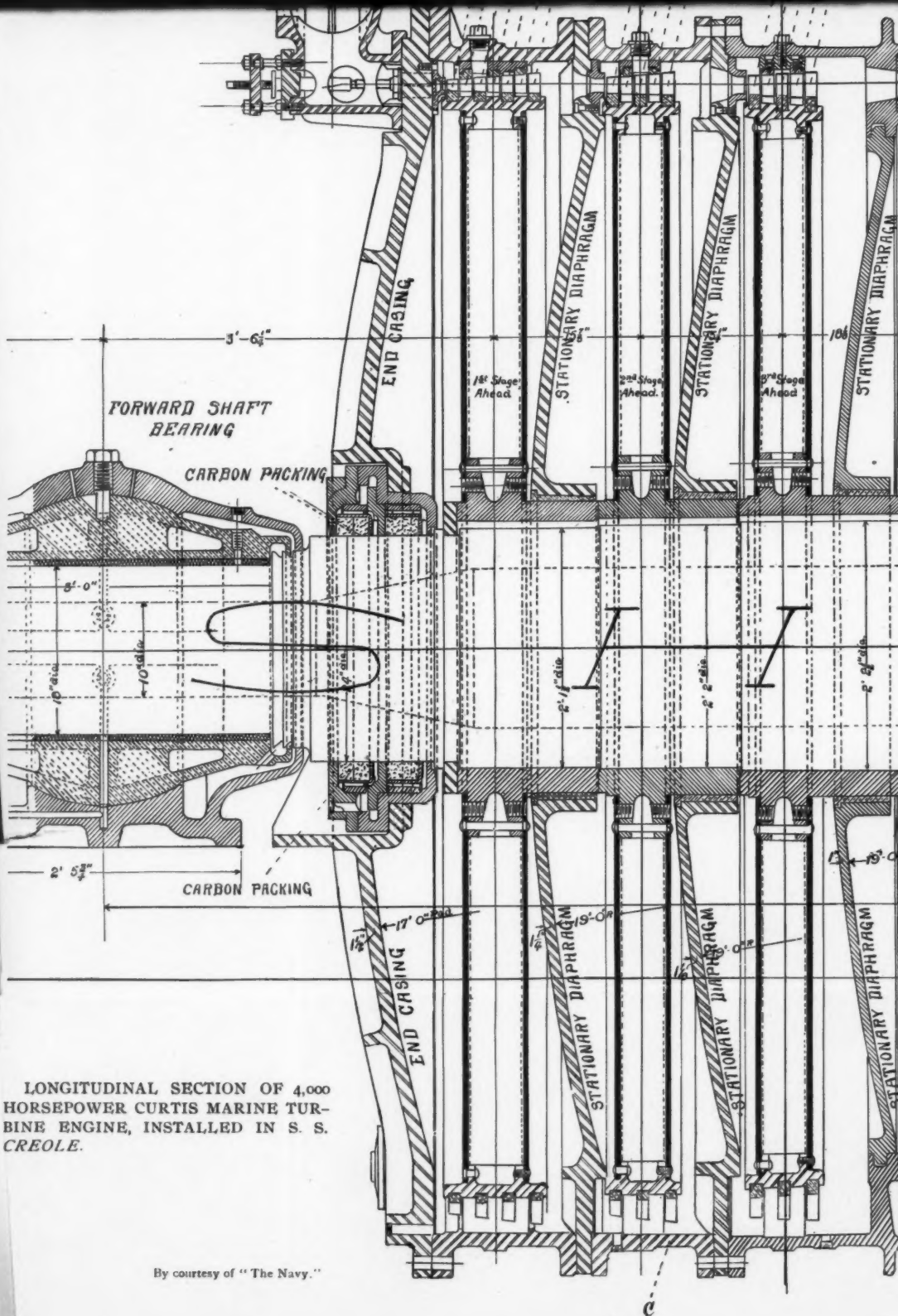
EXHAUST
STATIONARY DIAPHRAGM
3rd Stage Ahead
4th Stage Ahead
5th Stage Ahead
6th Stage Ahead
7th Stage Ahead
2'-1 1/8"
9-10 2nd
18'-6"



A Moving blades (buckets, or vanes), attached to circumference of moving wheels or rotors.

B Stationary blades (buckets, or vanes), attached to the inside circumference of the casing.

C Casing, which encloses the moving wheels or rotors and to which is attached the stationary diaphragms that separate one rotor from another and in the upper part of whose curves are put the nozzles, or "throttles" which direct the steam that has acted upon one rotor, against the moving blades of the succeeding rotor.



LONGITUDINAL SECTION OF 4,000 HORSEPOWER CURTIS MARINE TURBINE ENGINE, INSTALLED IN S. S. *CREOLE*.

By courtesy of "The Navy."

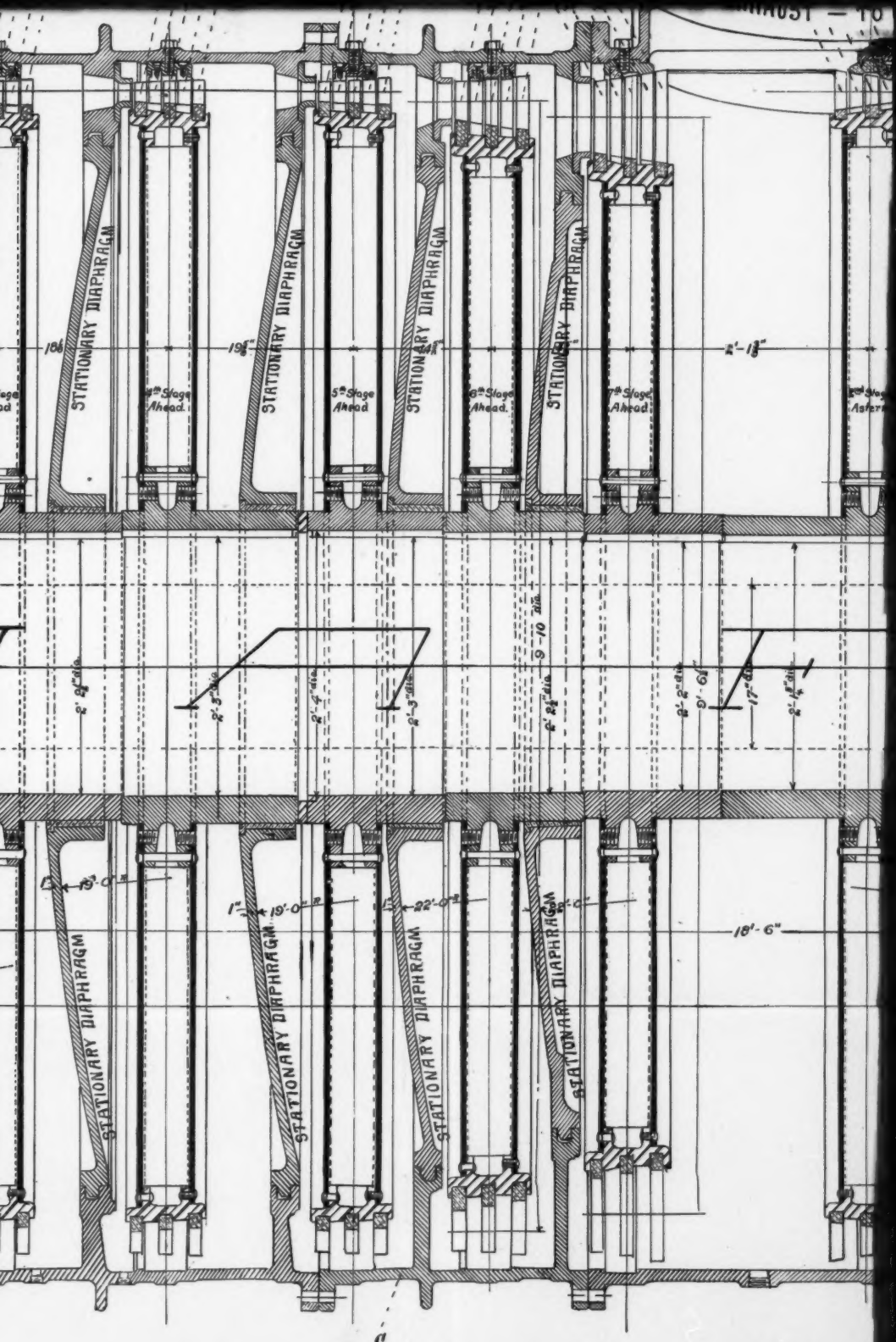


Fig. 5.—Vertical Section of "Creole's" Turbine.

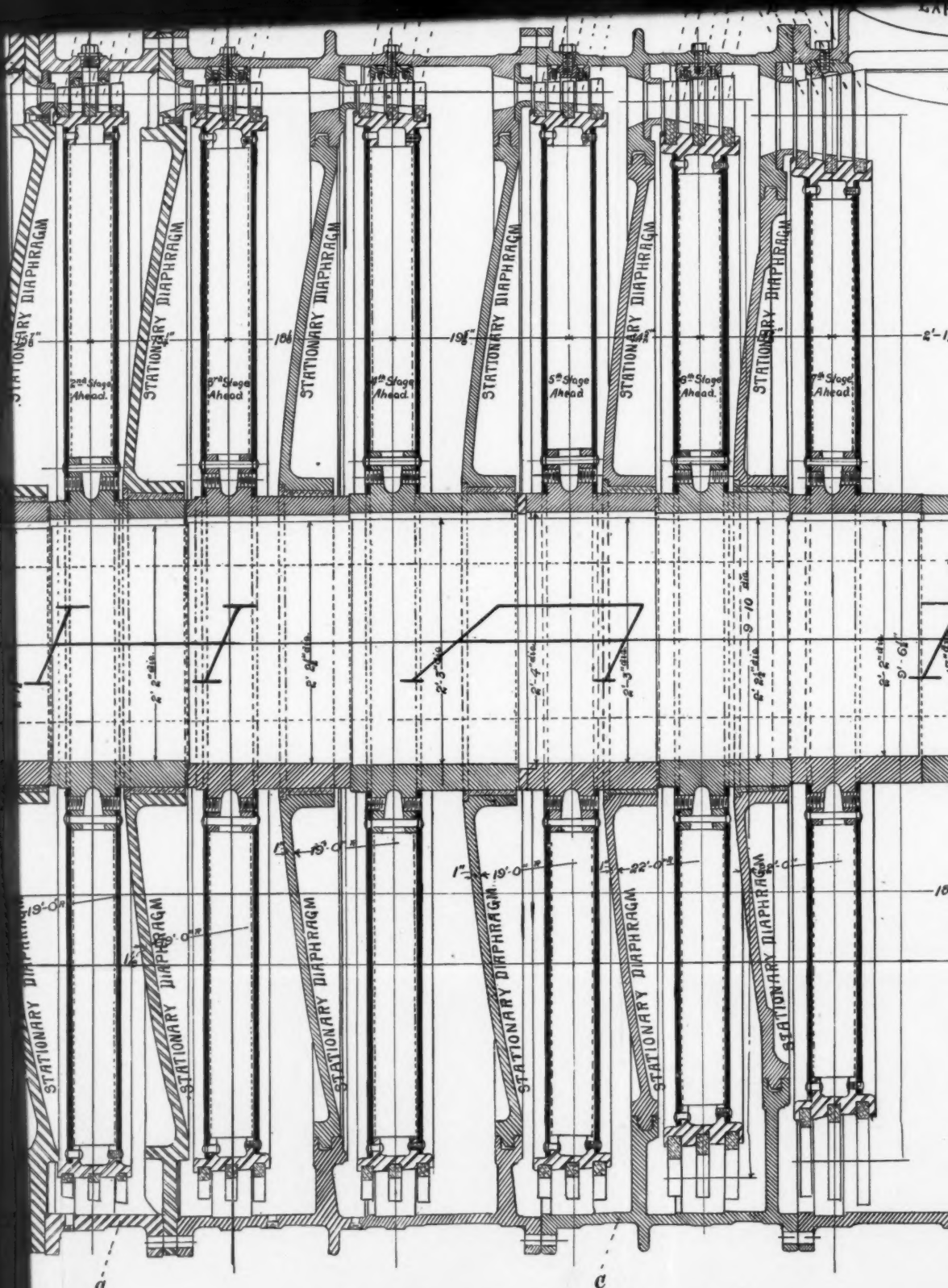
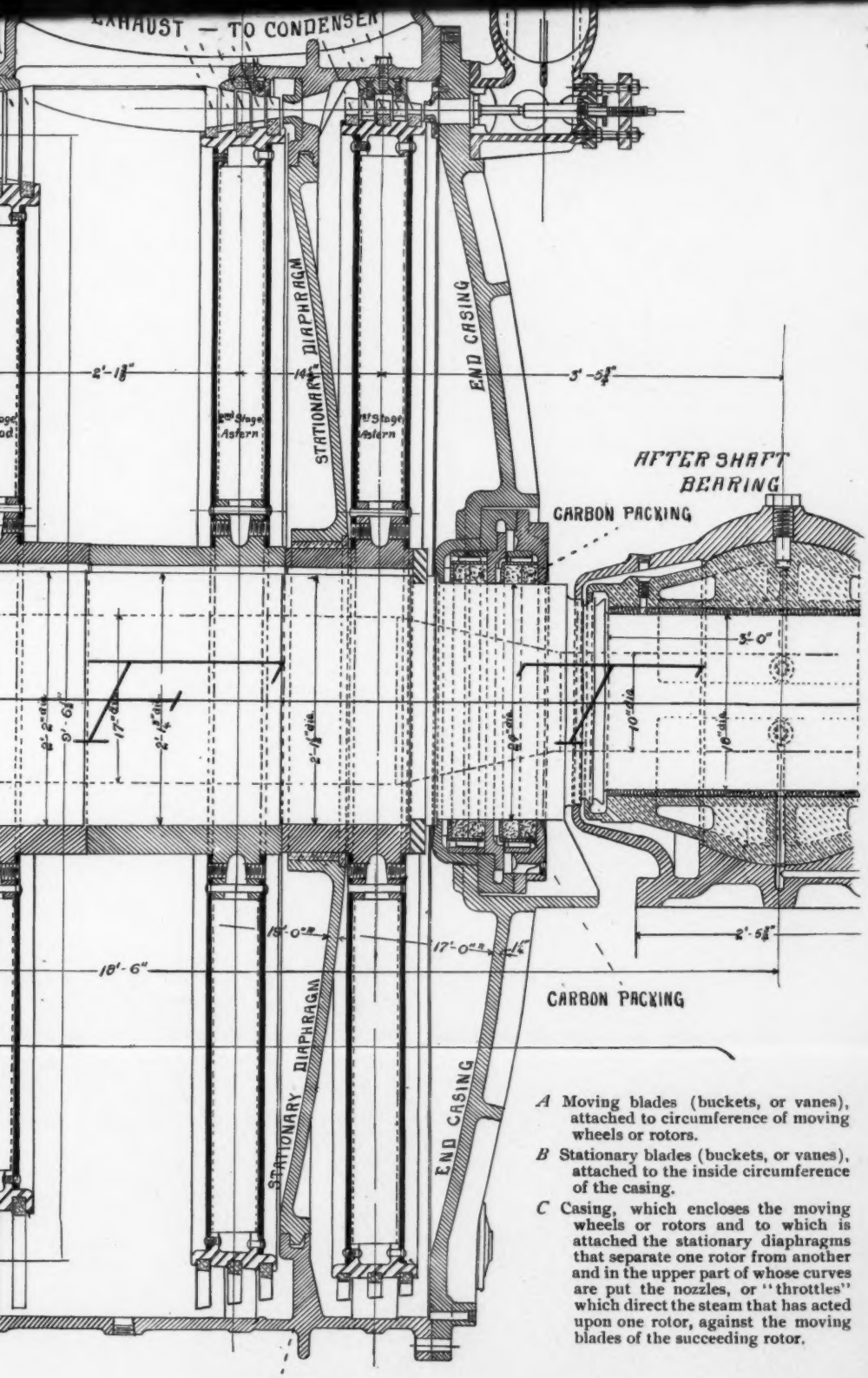


Fig. 5.—Vertical Section of "Creole's" Turbine.



A Moving blades (buckets, or vanes), attached to circumference of moving wheels or rotors.

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C Casing, which encloses the moving wheels or rotors and to which is attached the stationary diaphragms that separate one rotor from another and in the upper part of whose curves are put the nozzles, or "throttles" which direct the steam that has acted upon one rotor, against the moving blades of the succeeding rotor.

c

aligning bearings as shown. Where the shaft passes through the diaphragms they are provided with bronze bushings having a small clearance, thus preventing appreciable steam leak from one stage to the other. Where the shaft passes out through the ends of the casing it is provided with carbon stuffing boxes which, prevent steam leaking out at the ahead end or air leaking in at the back end where a vacuum exists.

The stuffing boxes are supplied with steam in the space between the carbon packing to prevent air leaking in and lowering the vacuum. They are also drained to the fourth-stage shell.

Cast-steel steam chests for ahead and astern running are attached to the front and back casing heads as shown, and are flanged for the main steam pipes. The nozzles for each stage are bolted to the diaphragms as shown, the diaphragms having steam-port openings cast in them to allow the steam to pass through to the nozzles.

Maneuvering is accomplished by means of two lever-operated balanced throttle valves, each taking steam from the main steam pipe, and one delivering to the ahead steam chest and the other to the astern steam chest. There are seven ahead wheels and two reverse wheels. The reverse wheels are mounted in the after end of the casing, and under ordinary ahead running they are in a vacuum and therefore do not waste power by steam friction. They are the same as the ahead wheels, except the blades are reversed. To reverse when going ahead the ahead throttle valve is shut and the reverse throttle valve opened, which is easily and quickly accomplished by the operating levers of the two throttle valves.

In the ahead steam chest there are twelve valves, each opening one of the nozzles for the first-stage wheel. For continuous running sufficient of the nozzles are opened to give the speed desired, and the ahead throttle valve is left open, thus giving full pressure in the steam chest. Nine nozzles will give full power, leaving three for overload. The astern steam chest has the same number of nozzles, but only six have valves.

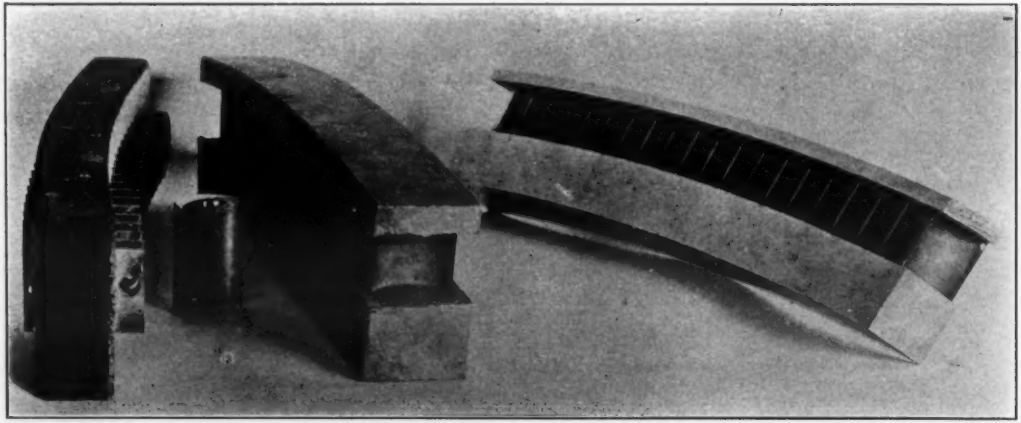
For maneuvering, the nozzle valves in the steam chests are left open and the speed is controlled by the throttle valves.

On each of the other six ahead stages two nozzle valves are fitted. Each valve closes one-fifth of the nozzle area of its stage. These valves are closed when running continuously at reduced speed, so that the proper steam pressure can be maintained in each stage.

Drain pipes are provided connecting each stage with the next, so the condensed steam in any stage will pass to the next one of lower pressure and there give up a part of its heat to do useful work. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. A regular marine thrust bearing is attached to the forward end of the turbine shaft. In addition to taking the propeller thrust this bearing also maintains the proper axial position of the rotor, so that the axial clearance of the blades is correct. This clearance is one-tenth of an inch on the first wheel and increases to one-quarter of an inch on the seventh wheel. The thrust is put at the forward end, so that any unequal expansion of the shaft and casing will be allowed for at the aft end where the clearance is largest. This axial clearance is very ample to allow for all unequal heat expansion that may occur and any mechanical irregularities and leave sufficient leeway for adjustment.

The construction of the blading is shown in Figure 6. The blades are made of special bronze accurately formed to the required shape by being extruded through a die and thus made into long bars. These are cut into lengths as required by each stage, and each length is notched or drilled through at the ends as shown at B. A number of blades are built up into a segment by casting on a composition base on the inner ends and a shroud on the outer ends as shown at C. In order to do this casting the blades are held in a sand core, with the ends projecting slightly as shown at A. After casting, the base and shroud are milled off true, thus completing the segment as shown at D.

The ends of the blades fuse into the cast parts, thus mak-



A B C D

By courtesy of "The Navy."

Fig. 6.—BLADING SEGMENT.

A Blades held in core previous to casting on base and shroud.
B Blade cut to length and notched.
C Segment after casting on base and shroud.
D Segment machined to size.

Fig. 7.—Calking in Blade Segments.

ing practically one solid piece of the entire segment. If any blade should not happen to fuse, the metal will flow into the notch or hole and effectually keep it in place. The extruding process gives the blades a hard, smooth skin, which reduces steam friction and effectually prevents wear and erosion of the blades.

The segments are held in the wheel rims by inserting the bases in rectangular grooves around the steel rims and calking the edges of the grooves. This calking is done by means of a pneumatic calking tool mounted on special adjustable pedestal, while the wheel is slowly revolved, as clearly shown in Figure 7. On the segments actually used in this vessel the shrouds were riveted on instead of being cast like the bases, but on the next turbines constructed they are cast as above described.

The bases and shrouds are overhung slightly beyond the edges of the buckets, so that if by any accident the wheels should be moved axially the blades would not come in contact and be damaged, but the overhanging bases and shrouds would protect them. These bases and shrouds are very strong, and can stand forcing together so hard that the turbine would be brought to a stop without any serious damage occurring. This very important feature renders stripping of the blades a practical impossibility in this design of turbine.

Fig. 7.—Calking in Blade Segments.

As before stated, the radial clearance at the ends of the blades can be made as large as desired without having any effect on the operation, and, as constructed, it varies from $\frac{1}{4}$ inch to 2 inches, depending upon the proximity of the stationary blade holders, as is clearly shown in Figure 5.

To allow for the increased volume of the steam as it expands in passing from stage to stage at lowering pressures, the length of the blades are increased, as shown in Figure 5, and also the arc of the nozzles is increased, thus giving greater area of passage in each succeeding stage. Also, in any *one* stage, the blade lengths are increased in each succeeding row, because the velocity falls as the steam passes from row to row, although it is at practically constant pressure throughout the stage.

In order to keep the pressure in the shell as low as possible, the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the firsts tage, and one-eighth in each of the other stages. This requires the first-stage nozzle to be of the expanding type, but all the other nozzles are of the parallel-flow type. Also the first-stage wheel is provided with four rows of buckets instead of three, as on the other wheels, since the greater energy drop produces greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. This arrangement makes all the ahead wheels except the first operate under eight-stage conditions.

PROPELLERS.

On the first trial variable-pitch propellers were used, but the results were not satisfactory. These were changed to two true-screw propellers, each having the following dimensions:

Number of blades	3
Diameter, feet and inches	10–9
Pitch, feet and inches	8–2
Pitch ratio	.76
Projected area, square feet	33.78
Developed area, square feet	36.0
Disc area, square feet	90.1
Immersion of tip at 24 feet 10 inches draught, feet and inches	11–5½

MEASUREMENT OF TRIAL DATA.

The brake horsepower of the port turbine was measured by a Föttinger torsion meter, which was calibrated in the shop on the section of shafting on which it was used.

The steam flow of the port turbine was measured by two tanks which received the discharge of the port air pump alternately and which emptied into the port hotwell. These tanks were calibrated by weighing the amount of water which each held.

The temperature of the steam in the turbine stages and steam chests was measured by high-grade glass thermometers inserted in mercury cups let into the steam spaces. The other

temperatures were measured by the regular ship thermometers which were permanently installed.

Pressures in the three lower stages and in the condensers were measured by mercury columns. Other pressures were

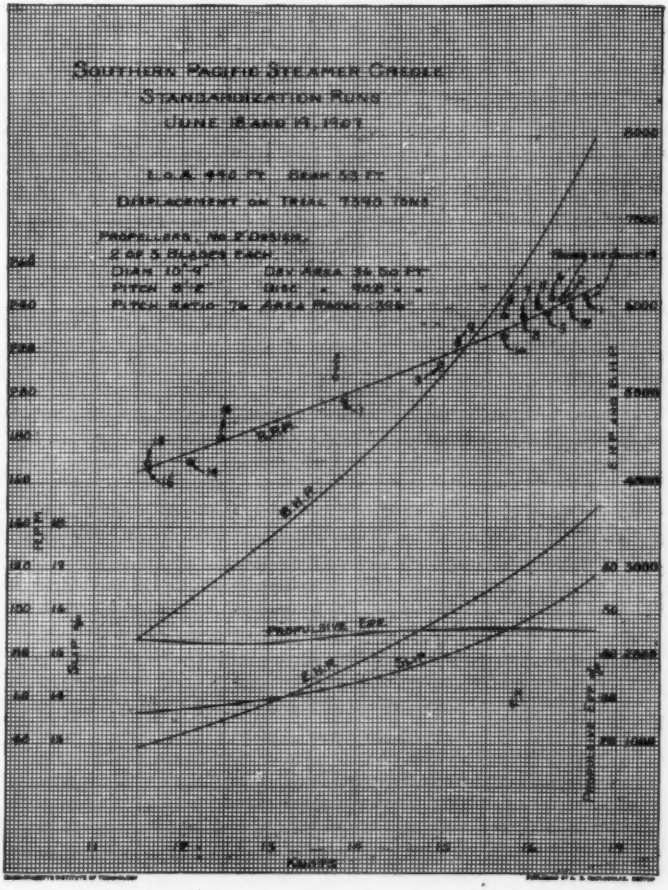


Fig. 8.

measured by the regular ship's gauges, which were kept adjusted for their working pressures by a weighted gauge tester.

Revolutions were taken from the ship's continuous counters.

The time on the measured mile was taken by two observers with stop watches, and upon crossing the line at both ends an electric signal was sent to the engine room to read the counters to obtain the revolutions on each mile run.

Torsion-meter and counter readings on the port turbine were read every minute, other engine-room readings every five minutes, and boiler observations every ten minutes.

Coal was measured by weighing the passing buckets as they entered the firerooms. On May 1st and June 19th each individual basket was weighed, but on the other runs the baskets were counted and the average net weight as before determined was used.

RESULTS.

STANDARDIZATION.

The standardization results of the runs over the measured mile, with the propellers finally used, are given on accompanying curve sheet, Figure 8.

It is to be noted that the propulsive efficiency obtained by the propeller used is very poor, although it is better than that obtained by the first propeller fitted to the vessel. A third design is now being built, and it is expected that a material improvement will be obtained, which, of course, will give a higher speed of the vessel for the same brake horsepower of the turbine.

FULL-SPEED RUN.

A four-hour run at full speed was made in June 19th, 1907, and the results given in Table 1. An average speed of 16.7 knots was obtained.

REVERSING TEST.

On two occasions the turbines were suddenly reversed with the vessel going at full speed ahead. The results of the test were made on June 18th are given in Table 2. It is seen that the vessel can be handled with remarkable quickness. Also she maneuvered perfectly when being taken through the drawbridge and in the narrow and crooked channel from Fore River. The vessel was docked at the Pier in North River without the assistance of tugs.

ECONOMY TESTS.

Steam consumption was measured on the port turbine on several runs, and the results are given in Table 3. During the run on May 17th the steam and vacuum conditions were maintained the best and most constant, and, therefore, the best water rate was obtained. The water rates obtained, namely, about 16 pounds per brake horsepower per hour for best steam and vacuum conditions, and about 17 pounds for average conditions, are about those to be expected of a triple-expansion reciprocating engine; suitable allowance being made for engine friction to compare these figures with *indicated horsepower* on which reciprocating-engine performances are based. In view of the fact that the power and speed of this vessel are low for her size, thus necessitating a bucket speed much below that giving maximum economy, these results are considered very satisfactory. The economy results also agree within a few per cent. with the performance of the experimental demonstration turbine from which the present design was developed, thus showing that results obtained on a small machine may be expected to be duplicated with confidence on a similar larger design.

By taking the power of the starboard turbine as equal to that of the port at the same revolutions, the total power of the vessel is obtained, and the economy results for both turbines on the run on May 17th are given in Table 4.

The results of boiler performance are given in Table 5.

Table No. 1.—FOUR-HOUR FULL-SPEED RUN.

Run was from Wood End Light around Boston Light Ship and return, on June 19th, 1907.

Down:	Distance by chart, knots	31.5
	Elapsed time, hours, minutes and seconds	1·52·20
	Average speed, knots	16.83
Back:	Distance by chart, knots	31.1
	Elapsed time, hours, minutes and seconds	1·52·25
	Average speed, knots	16.6
Total Run:	Average speed, knots	16.71

Table No. 2.—BACKING TEST.

Maximum power ahead = 247 R.P.M. 3,790 B.H.P. 7.17 Torsion meter for port turbine.

Time.	Chest steam pressure, gauge.	Vacuum in exhaust shell.	R.P.M.	Torsion meter reading.	B.H.P.	Per cent. of full torque.	Per cent. of full power.
1'10	...	...	...	4.0	...	55.8	...
1'11	200.	9	...	3.0	...	41.9	...
1'12	192.	9	135	2.95	854	41 2	22.5
1'13	187.	9	132	3.1	877	43.3	23.1
1'14	182.	13	127	3.05	870	42.6	22.9
1'15	175.	13	132	3.05	863	42.6	22.8
1'16	169.	14	136	2.95	860	41.2	22.7
1'17	165.	...	140	2.85	861	39.8	22.7
1'18	...	...	...	2.95	...	41.2	...

Time from throwing telegraphs till vessel came to a dead stop, minutes and seconds ..... 2'42

Distance traveled, lengths..... ..... 3½ to 4½

Time from throwing telegraph till reverse valves on turbines were opened, seconds, port 5

starboard..... 8

Time from opening reverse valve till turbines stopped, seconds, port..... 11

starboard..... 15

NOTE.—Low vacuum is caused by carbon packing in ahead and not being properly arranged to take reverse pressure.

Table No. 4.—ECONOMY RESULTS FOR BOTH TURBINES FOR RUN ON MAY 17, 1907.

Item.	9'40 to 10'00	10'00 to 10'30	10'30 to 11 00	11'00 to 11'30	11'30 to 11'50
R.P.M., port.....	236.5	239.6	244 8	250.8	249.7
R.P.M , starboard.....	228.4	232.8	239.7	242.2	240.6
Brake horsepower, port.....	3,450	3,585	3,805	3,950	3,920
starboard, est.....	3,030	3,195	3,480	3,595	3,520
total.....	6,484	6,780	7,285	7,545	7,440
Steam flow per hour, port.....	58,950	59,260	62,200	63,500	63,200
starboard, est.....	58,400	58,700	61,500	62,850	62,550
auxiliaries.....	10,600	10,600	10,600	10,600	10,600
total.....	127,950	128,560	134,350	136,950	136,350
Water rate, all purposes.....	19.75	18.95	18.45	18.16	18.35
Dry coal per B.H.P. per hour, all purposes...	2.28	2.19	2.13	2.1	2.12
port turbine only.	1.98	1.91	1.89	1.86	1.86

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Date.	June 19, A. M.									June 19, P. M.			
Length of run	8·00 to 8·30	8·30 to 9·00	9·00 to 9·30	9·30 to 10·00	10·00 to 10·30	10·30 to 11·00	11·00 to 11·30	11·30 to 12·00	12·00 to 12·30	7·30 to 8·00	8·00 to 8·30	8·30 to 9·00	9
Number of nozzles open	8	8	8 and 9	9	9	9	9	9	9 and 8	5	4	4	
Steam chest pressure, abs.	230	229.2	246	245.7	247.9	247.7	248	249.1	246.9	242.2	244.7	244.7	
superheat of	55.1	55.3	55.2	54.3	64	64.4	66.6	66.6	65	60.3	57.2	58.2	
Exhaust shell pressure, abs.	1.628	1.686	1.481	1.49	1.47	1.397	1.425	1.45	1.495	1.51	1.53	1.50	
Available energy B.T.U.	332	330	342.5	341.3	345.5	348	347	347	344.5	341.3	340.5	342.2	
R.P.M.	228.9	228	240.8	245.5	245	247.3	247	247.4	243.1	192.5	178	177.4	
Torsion meter reading	5.9	5.83	6.68	7.00	7.13	7.16	7.17	7.17	6.98	4.14	3.57	3.61	
Brake H.P.	2,890	2,843	3,440	3,680	3,740	3,790	3,790	3,795	3,630	1,706	1,360	1,370	1
Steam flow, pounds per hour	54,000	53,600	60,250	64,100	63,700	63,400	64,600	64,000	62,950	37,500	31,660	30,250	30
W.R. per B.H.P., turbine only	18.69	18.85	17.52	17.43	17.03	16.73	17.05	16.87	17.34	21.98	23.28	22.08	
Efficiency, per cent.	41.0	40.9	42.4	42.8	43.2	43.7	43	43.5	42.6	34.0	32.1	33.7	
Efficiency of demonstration, machine same speed	40.7	40.5	42.0	42.6	42.5	42.7	42.7	42.7	42.2	35 8	33.7	33 6	
Creole compared to demonstration, per cent. difference	+0.8	+1.0	+1.0	+0.5	+1.7	+2.3	+0.7	+1.9	+1.0	—5.04	—4.74	+0.3	
W.R. per B.H.P., *Creole*, referred to 27½-in. vacuum	...	...	...	16.92	16.64	16.45	16.74	16.54	16.85	...	...	...	

Table 3.—SUMMARY OF PORT TURBINE.

June 19, P. M.				June 20.									
8·00 to 8·30	8·30 to 9·00	9·00 to 9·30	9·30 to 10·00	8·30 to 9·00	9·00 to 9·30	9·30 to 10·00	10·00 to 10·30	10·30 to 11·00	11·00 to 11·30	11·30 to 12·00	12·00 to 12·30	12·30 to 1·00	1·00 to 1·30
4	4	4	4	7	7	7	7	8	8	8	8	8	8
244·7	244·7	247·2	248·2	244·2	244·7	245·7	238·7	237·7	240·7	241·7	245·2	238·3	244·1
57·2	58·2	56·2	57·6	65·7	57·1	62·8	72·3	72·7	70·4	69·2	69·5	70·6	72·9
1·53	1·50	1·50	1·50	1·46	1·374	1·323	1·20	1·372	1·215	1·298	1·176	1·176	1·26
340·5	342·2	342·5	342·8	345 5	350·0	349·5	354·3	348·0	354·7	351·5	357·0	355·0	354·3
178	177·4	177·8	177·7	221·7	220	220·8	221·2	232·4	233·6	232·1	233·1	230·6	233·8
3·57	3·61	3·62	3·66	5·64	5·70	5·67	5·56	6·32	6·93	6·26	6·3	6·12	6·89
1,360	1,370	1,376	1,391	2,674	2,730	2,680	...	3,145	3,113	3,110	3,140	3,000	3,142
31,660	30,250	30,580	32,750	47,500	49,700	48,900	...	760,000	55,500	55,000	56,100	54,600	56,000
23.28	22.08	22.2	23.52	17.75	18.2	18.25	...	19.08	17.82	17.68	17.88	18.09	17.82
32.1	33.7	33.45	31.6	41.5	40.0	39.9	...	38.4	40.2	41.0	39.9	39.7	40.25
33.7	33 6	33.7	33.7	37.7	39.5	39.6	...	41.0	41.2	41.0	41.0	40.9	41.2
−4.74	+0.3	−0.74	−6.24	+4.6	+1.3	+0 7	...	−3.8	−2.5	...	−2.9	−2.94	−2.31
...	...	...	...	...	...	...	...	...	...	...	...	...	...

Table 3.—SUMMARY OF PORT TURBINE.

19, A. M.					June 19, P. M.										
9·00 to 10·30	10·30 to 11·00	11·00 to 11·30	11·30 to 12·00	12·00 to 12·30	7·30 to 8·00	8·00 to 8·30	8·30 to 9·00	9·00 to 9·30	9·30 to 10·00	8·30 to 9·00	9·00 to 9·30	9·30 to 10·00	10·00 to 10·30	10·30 to 11·00	11·00 to 11·3
9	9	9	9	9 and 8	5	4	4	4	4	7	7	7	7	8	8
247.9	247.7	248	249.1	246.9	242.2	244.7	244.7	247.2	248.2	244.2	244.7	245.7	238.7	237.7	240.7
64	64.4	66.6	66.6	65	60.3	57.2	58.2	56.2	57.6	65.7	57.1	62.8	72.3	72.7	70.4
1.47	1.397	1.425	1.45	1.495	1.51	1.53	1.50	1.50	1.50	1.46	1.274	1.323	1.20	1.372	1.2
345·5	348	347	347	344·5	341·3	340·5	342·2	342·5	342·8	345 5	350·0	349·5	354·3	348·0	354·7
245	247·3	247	247·4	243·1	192·5	178	177·4	177·8	177·7	221·7	220	220·8	221·2	232·4	233·6
7·13	7.16	7.17	7.17	6.98	4·14	3·57	3.61	3.62	3.66	5.64	5.70	5.67	5.56	6.32	6.2
,740	3,790	3,790	3,795	3,630	1,706	1,360	1,370	1,376	1,391	2,674	2,730	2,680	...	3,145	3,113
,700	63,400	64,600	64,000	62,950	37,500	31,660	30,250	30,580	32,750	47,500	49,700	48,900	...	?60,000	55,500
17.03	16.73	17.05	16.87	17.34	21.98	23.28	22.08	22.2	23.52	17.75	18.2	18.25	...	19.08	17.8
43·2	43·7	43	43·5	42.6	34·0	32·1	33·7	33·45	31.6	41.5	40.0	39.9	...	38.4	40.2
42·5	42.7	42.7	42.7	42.2	35 8	33·7	33 6	33·7	33·7	37·7	39·5	39.6	...	41.0	41.2
+1.7	+2.3	+0.7	+1.9	+1.0	−5.04	−4.74	+0.3	−0.74	−6.24	+4.6	+1.3	+0 7	...	−3.8	−2.5
16.64	16.45	16.74	16.54	16.85	...	...	...	...	...	...	...	...	...	...	...

June 20.										May 17.			
10°30 to 11°00	11°00 to 11°30	11°30 to 12°00	12°00 to 12°30	12°30 to 1°00	1°00 to 1°30	1°30 to 2°00	2°00 to 2°30	2°30 to 3°00	3°00 to 3°30	10°00 to 10°30	10°30 to 11 00	11°00 to 11°30	11°30 to 11°50
8	8	8	8	8	8	8	8	8	8	9	9	9	9
237.7	240.7	241.7	245.2	238.3	244.1	247.7	247.8	245.7	239.2	239.1	248.6	252.7	251.3
72.7	70.4	69.2	69.5	70.6	72.9	54.6	69	70.8	65.7	71.9	73.5	74.5	74.9
1.379	1.215	1.298	1.176	1.176	1.26	1.334	1.372	1.455	1.642	1.375	1.20	1.177	1.105
348.0	354.7	351.5	357.0	355.0	354.3	347.7	349.5	346.7	337.4	349.0	357.8	360.5	362.5
232.4	233.6	232.1	233.1	230.6	233.8	235	234.9	231.6	227.4	239.6	244.8	250.8	249.7
6.32	6.23	6.26	6.3	6.12	6.29	6.34	6.35	6.31	6.09	6.99	7.26	7.35	7.33
3,145	3,113	3,110	3,140	3,020	3,142	3,190	3,193	3,130	2,930	3,585	3,805	3,950	3,920
60,000	55,500	55,000	56,100	54,600	56,000	56,900	57,000	56,700	54,200	59,960	62,200	63,500	63,200
19.08	17.82	17.68	17.88	18.09	17.82	17.83	17.84	18.1	18.5	16.52	16 34	16.06	16.12
38.4	40.2	41.0	39.9	39.7	40.25	41.0	40.75	40.6	40.75	44.05	43.45	44.0	43.55
41.0	41.2	41.0	41.0	40.9	41.2	41.3	41.3	41.0	40.5	41.6	42.2	42.8	42.7
—3.8	—2.5	...	—2.9	—2.94	—2.31	—0.7	—1.33	—0.97	+0.62	+5.9	+3.0	+2.8	+2.0
...	...	...	...	...	...	...	...	...	...	...	...	...	...

Table 5.—BOILER RESULTS.

	May 17.	June 19.
Date		
Duration	9·45 A. M. to 11·50 A. M.	8·30 A. M. to 12·30 P. M.
Boiler pressure, pounds, absolute	262.9	262.0
superheat, degrees Fahrenheit	82.0	56.0
Feed-water temperature, degrees Fahrenheit	192.0	186.0
B.T.U. in steam	1,259.0	1,242.0
Factor of evaporation	1.139	1.128
Water evaporated per hour, actual	132,800.0	132,964.0
from and at 212 degrees Fahrenheit	151,300.0	150,000.0
Wet coal burned per hour	15,800.0	15,361.0
Dry coal burned per hour	15,340.0	15,020.0
Combustible burned per hour	13,530.0	13,770.0
B.T.U. per pound of dry coal	13,880.0	14,653.0
Actual evaporation per pound of wet coal	8.40	8.67
dry coal	8.65	8.85
combustible	9.81	9.65
Evaporation from and at 212 degrees Fahrenheit, per pound of wet coal	9.58	9.67
dry coal	9.87	10.0
combustible	11.2	10.89
Boiler efficiency, per cent	68.5	65.7
Dry coal per square foot of grate area, per hour	19.9	19.5
Water evaporated per square foot boiler heating surface, actual, per hour	5.48	5.48
Water evaporated per square foot total heating surface, from and at 212 degrees Fahrenheit, per hour	5.3	5.25
Tons of coal burned per hour, actual	7.06	6.87

NOTES.

TAR OILS FOR DIESEL ENGINES.

When the Diesel engines were first introduced it was thought that the great future of the new motors rested on the fact that an engine had been brought out which would consume any kind of liquid fuel. As is often the case, the development took place on rather different lines. The Diesel engine showed its own preferences as regards fuel, just like the gasoline motor, and other difficulties had first to receive attention and be overcome. This fuel problem is not serious in our country. A Diesel motor is fed with so-called paraffin, which can be obtained at very cheap rates, and motors of this type are more commonly in use here than on the Continent, and in Germany, whence they hail, and where import duties keep the fuel question alive. A discussion of these features is, unfortunately, complicated by the disgraceful state of the nomenclature concerning oils. The engineer and the scientific chemist are sure to misunderstand one another in such a discussion, and it seems high time to put a stop to this confusion.

In Germany tar oils are broadly distinguished as lignite-tar oils and coal-tar oils. The tar oils gained from the distillation of wood and peat are not at present of much importance. Again, speaking broadly, lignite-tar oils are considered a suitable fuel for Diesel engines, while most of the coal-tar oils are regarded as unsuitable. The coal-tar oils that are sometimes utilized in Diesel motors are creosote oil and anthracene oil, whose boiling points lie above 240 degrees centigrade. These oils are very much cheaper than the lignite oils; they can be obtained in Germany for from 25s. to 50s. per 100 kilogrammes, while the lignite oils command prices ranging from

80s. to 110s. The low price of the former product is partly due to some undesirable qualities of these oils: bad smell, want of uniformity, &c. But as the German production of coal-tar oils is now double that of lignite oils, the utilization of the cheaper fuel in Diesel motors is a question of considerable interest. Of the lignite-tar oils the less expensive grades, which do not contain much paraffin proper—*i. e.*, hydrocarbons of the C_nH_{2n+2} series, valuable as wax—are generally used in Diesel motors; the density of those oils lies between 0.860 and 0.926.

Although there is no inducement to use a high-grade paraffin oil as fuel, general experience seems to have suggested that a tar oil containing more than 2.5 per cent. of paraffin would not answer in Diesel motors. Mr. Paul Rieppel, of the Vereinigte Maschinenfabrik Augsburg and Maschinenbau-Geselleshaft Nürnberg, found, however, that a higher percentage of paraffin did not in itself disqualify a tar oil as Diesel-motor fuel. He raised the paraffin proportion by steps to 15.2 per cent., and worked motors with such fuels for more than fifty hours without experiencing any trouble. As, moreover, crude petroleum and gas oils of foreign origin, containing considerably more than 2.5 per cent. of paraffin, answer perfectly in Diesel engines, the conclusion that the paraffin percentage was not, as such, a criterion for the suitability of a Diesel-motor fuel appeared to be justified. Mr. Rieppel thereupon resolved to investigate the whole problem with the object of ascertaining which oils and mixtures can be recommended and the cause of their suitability. The experiments have been carried out in the engineering laboratories of the firm at Augsburg and Nuremberg, in the chemical laboratory of the Gewerbe Museum at Nuremberg, in the engineering laboratory of the Technical High School at Charlottenburg, and in the Institut für Gährungsgewerbe (fermentation) in Berlin. Though the subject has not been exhausted, and investigation will be carried further, some important conclusions can already be drawn from Mr. Rieppel's account of his research in the "Zeitschrift des Vereins Deutscher Ingenieure."

We may also draw attention to a memoir on liquid fuels for internal-combustion motors, published by Mr. Kutzbach, a colleague of Mr. Rieppel, in previous numbers of the same journal.

As a rule, a 35-horsepower motor was worked under normal conditions in the thirty-eight series of experiments made on twenty-one oils and mixtures. All these materials were tested chemically and physically, and these latter tests were extended, for the sake of comparison, to some oils, of which sufficient quantities could not be obtained for long trials. The number of oils investigated thus comes up to twenty-six. One of the first materials tested was benzine. It proved very unsatisfactory. There were misfires and violent explosions, the exhaust would shoot out as a column of fire and soot, and pistons and valves became badly tarred. When the load was reduced below normal, the fuel would not ignite at all. Mixtures of benzine with tar-oils, especially the creosote and anthracene oil mentioned, worked better; at full load up to 40 per cent. of the latter products could be added to benzine; but such mixtures are not really to be recommended.

The calorific values of the oils were obtained with Mahler bombs. For the determination of the viscosity, Engler's method was first applied, and the experiments were conducted at 20 degrees centigrade (68 degrees Fahrenheit). As the viscosity changes with the temperature, however, it was thought best to measure the viscosity at the temperature at which the oil is forced through the atomizer. These temperatures could not be determined directly—not even with the aid of thermo-couples—and the following indirect method was, therefore, resorted to: A 12-horsepower Diesel motor was fitted so that the atomizer could be removed 45 and 42 seconds after stopping the motor, and immediately be dropped into a calorimeter. The temperatures thus deduced for the atomizer were 74 and 76 degrees centigrade. These temperatures are lower than might be expected, since the atomizer projects partly into the combustion space; but there is cool-

ing, directly through the oil and air injected, and indirectly through the circulating water; and a temperature of 80 degrees centigrade (176 degrees Fahrenheit) was, therefore, adopted as the probable temperature of the oil at the moment when its viscosity has effect. So far as the experiments go, the viscosity is not a general criterion. A certain viscosity value may be most favorable for a particular kind of oil, but this viscosity cannot guide us as to other oils. Flash point, temperature of ignition and calorific values were also determined, but none of these features seem to be guiding characteristics for the suitability of an oil.

The elementary analysis, the determination of carbon and hydrogen and of their ratios, seemed to be a better criterion. The lignite-tar oils are mostly hydrocarbons of the fatty series corresponding to the formulae $C_n H_{2n}$ or $C_n H_{2n+2}$. In the coal-tar oils the aromatic hydrocarbons, benzine (C_6H_6), and its derivatives, predominate, and they are relatively poorer in hydrogen than the oils of the former class. The combustion analyses, made in Dennstedt electric furnaces with the aid of platinized quartz, proved that the ratio H : C in the good Diesel-motor oils ranged from 1.53 to 2; while in the coal-tar oils the value of this ratio was generally below 1. The anthracene and creosote-tar oils previously mentioned gave the ratios 0.82 and 0.81, and were thus unsatisfactory; by mixing them with paraffin oil the ratio can be raised to 1.39, however, and they then become fairly suitable. The distillation tests proved still more characteristic than the ratio H : C. The distillation, performed more conveniently in copper stills than in glass retorts, indicates what portion of the oil is volatilized or gasified at a certain temperature. When this distillation was watched it was seen that some oils would generate oil gas at a fairly uniform rate, while others would gasify irregularly in its fits and starts. The former oils are suitable, the latter not. The lignite oils showed, on the whole, a more uniform distillation then the coal-tar oils, and one understands why the former should constitute the better fuel. The Scottish tar oil, obtained from cannel coal and boghead shale,

ranked with the lignite oils, the gasification being quite uniform. It differed from the lignites in so far as the lignites are completely gasified at a temperature of 400 degrees centigrade, while the Scotch fuel leaves a pitch residue representing 43 per cent. of the fuel bulk. The formation of a pitch does not appear to be injurious in a Diesel motor, however, as the subsequent combustion is sufficiently intense to burn all such solid residues.

It was further of interest to study the behavior of the various fuels at high pressures and temperatures. The spherical bombs which Mr. Rieppel constructed for the purpose of getting complete combustion records did not answer, and he adopted a simpler bomb of his colleague, Mr. K. Kutzbach. This bomb is cylindrical, with a disc flame, the stem being formed of two concentric cylinders; the annular space is charged with the oil, the thermometer inserted in the inner cylinder, and the bomb closed by screwing down the cover, to which the pressure gauge is attached, to its flange. The annular space has a capacity of 75 cubic centimeters, and it is charged with about 30 cubic centimeters of the oil, which is heated by Bunsen burners. Control observations made with steam and benzine charges demonstrated that the pressures and temperatures were not recorded quite correctly, but the accuracy appeared sufficient for comparisons. Plotting temperatures as abscissae, and pressures as ordinates, the experimenters obtained curves which would first rise regularly, and then ascend steeply, indicating that the pressure first increased with the temperature, and that afterwards the further supply of heat did not cause any further rise of temperature, but increased the pressure, owing to decomposition and the generation of gas. The gas is, according to Kutzbach, a mixture of hydrogen and hydrocarbons, and it is said to be the hydrogen which first catches fire, a statement which appears to be disputable. In some cases the thermometer actually fell a little on further heating; this fall, if real, and not due to instrumental errors, might be ascribed to a dissociation of the gas. Crude petroleum gave a regular curve, indicating evapo-

ration without decomposition. Creosote and anthracene tar oils gave similar curves up to a certain point; then a lively gas generation set in. With Scotch liquid fuel and the lignite oils, this gas generation sets in at comparatively low temperature, and that circumstance would explain the suitability of these oils for Diesel motors. With coal-tar oils the gas generation becomes brisk only at higher temperatures, and they would not give sufficient gas for spontaneous ignition in the Diesel motor.

The difference in chemical constitution may account for this different behavior. The coal-tar oils are rich in derivatives of the benzine series, whose benzine nucleus is not easily split up, and such a splitting up would have to precede decomposition. The lignite oils contain essentially hydrocarbons of the fatty series, whose side chains are more easily split up; liquid fuel naturally ranks with the lignites, being a shale oil. We also understand why a high percentage in hydrogen should be desirable, though the hydrogen percentage does not directly decide the suitability of the oil as Diesel-engine fuel. The paraffins of the fatty series are rich in hydrogen, and the hydrogen in the resulting gas mixture burns readily. These conclusions are supported by the tests on carburetting oil which A. Spiegel has quite recently published ("Schilling's Journal für Gas und Wasser," 1907, No. 3); an oil makes a good carburettor if it is rich in paraffins.

One point, however, appears to be rather contradictory. Benzine was characterized as unsuitable for Diesel motors, while it answers perfectly in internal-combustion motors, and, being a uniform substance, it should give, and did give, a regular curve in the bomb. The discrepancy is explained by the different working conditions. In the gas motor the benzine vapor is drawn in, compressed by the next stroke, mixed with air, and the explosive mixture is ignited by an electric spark. In the Diesel engine the liquid fuel is injected into the highly-compressed air, in which it is at once to catch fire spontaneously; the whole process is much quicker. That the Diesel engine will not work well with

benzine is no drawback, of course. If the mixture could locally be heated to higher temperatures, or otherwise be made to burn spontaneously, the Diesel engine might work with the less combustible coal-tar oils. Mr. Rieppel has made very satisfactory laboratory experiments in this direction, and fed Diesel motors with coal-tar oils for weeks. The matter, however, remains, so far, in the experimental stage.—"Engineering."

CORDITE.

All the nitro-powders, being highly exothermic chemical compounds, in place of a mere mixture of reagents, such as common black powder, are much more liable than the latter to deterioration when stored under high temperatures. The disastrous explosion on board the *Jéna* will be in the minds of all, and our own Navy and Army have also suffered from this unfortunate characteristic of the modern type of propellant, though, fortunately, on a much less serious scale. In December, 1899, a 6-inch cartridge on the *Revenge* ignited spontaneously, and two other cartridges in the same box were also found to have gone off, and last October a similar incident took place on board the *Fox* whilst at Bombay. In the autumn of last year, moreover, there were two spontaneous explosions at cordite magazines in India. When it recently became known that the Home Office had laid an embargo on certain cordite manufactured by Messrs. Kynoch because it contained an unauthorized ingredient in the shape of mercuric chloride, it was not therefore surprising that a large section of the public jumped to the conclusion that these disasters arose from defective ammunition purchased from the firm in question, and the politicians with socialistic leanings rejoiced in a new argument for the abolition of the private contractor. Unfortunately for their thesis, this ammunition turns out in each case to have originated from the Government factories at Waltham, and there is thus every prospect that the nation will continue to reap the benefit of the ingenuity and enterprize of private firms.

It is impossible for any one individual to know everything. It is, therefore, most important that the labors of the Government experts, able as these are, should be supplemented by the work of independent experimenters. Indeed, most of the radical improvements in armaments, from the Maxim gun to the submarine boat, have originated in this fashion. Cordite itself is no exception, for though it was first compounded in the Government laboratories, the authorities frankly admitted that their investigations were based on the pioneer work of Nobel. Again, until the introduction of the so-called Palma Trophy ammunition by the King's Norton Company, the Lee-Enfield rifle proved inferior at long ranges to the Mannlicher and Mauser, whilst subsequently it held the advantage. It is, therefore, a matter for congratulation that the spontaneously-ignited ammunition cannot be represented as the natural outcome of a contractor's "greed for dividends," but was actually manufactured at Waltham, under the superintendence of officials as to whose complete honesty there is absolutely no question.

Nevertheless, although our national forces have suffered no disaster from the ammunition supplied by Messrs. Kynoch, the action of this firm in adding mercuric chloride to the cordite supplied by them appears to be indefensible. In itself the addition does no harm, as the official experts admit, and it is quite possible that Messrs. Kynoch in all honesty believed that the addition was actually an improvement. In such circumstances it is not uncommon for contractors to take matters into their own hands, incorrect as this may be from the standpoint of law and ethics. It is, for example, notorious that certain specifications for some special gunmetals for steam fittings specifically exclude the use of lead, yet all practical men know that a certain proportion of this is necessary if a sound casting is to be obtained. A small quantity is, therefore, quietly added by some foundrymen; but it seems, in certain cases at any rate, to entirely disappear in the operation of casting, although, before vanishing, it has performed its desired function of causing the tin and copper

to form a satisfactory alloy. Where the interests concerned are, however, so large as in the present instance, no firm, no matter how well assured they may themselves be as to the resulting improvement of their product, have the right to violate a clause of the specification drawn up by the experts held responsible by the nation for the quality of the ammunition supplied to our sea and land services.

The objection to the presence of mercuric chloride in cordite has in some cases been misunderstood. It is not, as Mr. Haldane very clearly pointed out in his admirable speech on Tuesday evening, that good cordite is injured by the addition of this ingredient. Of this there appears to be no evidence, and Messrs. Kynoch, no doubt, believe that it is actually improved by it; but the fact remains that the presence of even a minute quantity of this compound will enable a bad and dangerous sample to successfully pass through the ordinary official "heat" test.

In this test 20 grains of cordite are placed in a test tube, in the upper half of which is suspended a strip of filter paper which has been impregnated with a mixture of starch and potassium iodide. The upper half of this strip is moistened with a 50 per cent. solution of glycerine in water. The tube and its contents are then heated up in a water bath to 180 degrees Fahrenheit, and at the end of a certain time, more or less prolonged, according to the quality of the sample, a brown line makes its appearance at the junction of the dry and moist halves of the paper. With a good sample of the regulation cordite this line should not appear till after the test has lasted twenty minutes. The brown coloration is due to the production of nitrous fumes, which liberate the iodine from the potassium iodide. The addition of alkalies, urea, or, above all, mercuric chloride to the sample renders the test invalid, the nitrous fumes liberated combining with the addition, so that they consequently fail to reach the potassium iodide. This fact has long been known and has been recognized in the specifications governing the acceptance of nitropowders, not only of our own, but of other governments. The

regulations of the United States Army Ordnance Department, for instance, say that "the presence in a powder of mercuric chloride, or alkali, or any other substance which may in any way mask the heat test, will be sufficient to cause its rejection." The specifications of our own authorities are equally explicit, so that the action of Messrs. Kynoch in adding the mercury salt to the cordite supplied by them is, as already pointed out, quite indefensible. If persuaded that good cordite was made still better by the addition, their proper course was obviously to convince the authorities of this, and get them to amend the specification accordingly; but in adding without permission a reagent which, it is admitted, would enable dangerous samples of the explosive to pass the official test, they accepted, even on the most favorable interpretation of their action, a most serious and dangerous responsibility.

The authorities are evidently taking a sane and conservative course in the matter. Instead of destroying without investigation many tons of what may be perfectly good and safe cordite, they are carefully examining and testing the whole of the doubtful stock. Some extremists in the House have taken the view that the Government should be held immediately responsible for the consequences of any decision which may finally be come to in the matter. Politicians in opposition are peculiarly prone to this contention, but in technical and scientific matters the real responsibility must rest upon the experts of a department, and not on the mere political mouthpiece. The latter is only open to blame if, after procuring the best scientific advice available, he fails to follow it. In an analogous case, for instance, the responsibility for the success of a surgical operation must fall on the operator, and not on those who called him in.—"Engineering."

"WHITE-FORSTER" WATER-TUBE MARINE BOILERS.

The "White-Forster" patent water-tube marine boilers are well known and have been largely adopted, especially for torpedo-boat destroyers for the British and foreign navies, and also for other vessels. Machinery fitted with "White-Forster"

boilers, Fig. 1, shows the design of boiler as fitted to the destroyer H. M. S. *Ness*.

Fig. 1.

Several important improvements have recently been made, and the accompanying illustrations show the latest types. Fig. 2, shows a 2,000 I.H.P. boiler for H. M. S. *Cricket*, coastal destroyer.

The principal features of the "White-Forster" boiler will be readily understood from Figs. 1 and 2. From these it will be seen that there are two lower water drums connected by the generating tubes to an upper steam and water drum, the water level being arranged as to entirely drown all the tubes. The radius of curvature of each tube is the same, and the curvature is only sufficient to determine the direction of movement due to expansion, and also to facilitate cleaning and repairs. The tubes are arranged in position in a transverse section like staves in a section of a barrel, the continuation of the line of curvature of each tube, passing through the top drum in line with the end manhole, allows each tube to be inserted or withdrawn through this manhole, so that it is possible to withdraw any tube without disturbing the remaining tubes, and as every tube has the same curvature they can

be readily cleaned, internally, by a tube brush having a rigid handle curved to the same radius. The boiler and its casings are constructed entirely of mild steel, no castings of any kind being employed. Large downtake tubes are fitted. Ordinary manhole doors are fitted to all drums, studded doors being avoided. As the inclination of the generating tubes is considerable, and usually varies between 40 degrees and 60 degrees, the circulation is definite and rapid, keeping the tubes free from deposit. No special water baffles or separator plates are found necessary, the usual internal steam pipe being sufficient to give dry steam. Special care is taken in designing the casings and uptakes so as to make the boiler as efficient as possible. Patent baffles are fitted so as to divide the uptake into two or more parts, in such a manner that the gases are drawn equally over the whole tube surface, and at right angles to it, without offering any resistance or increas-

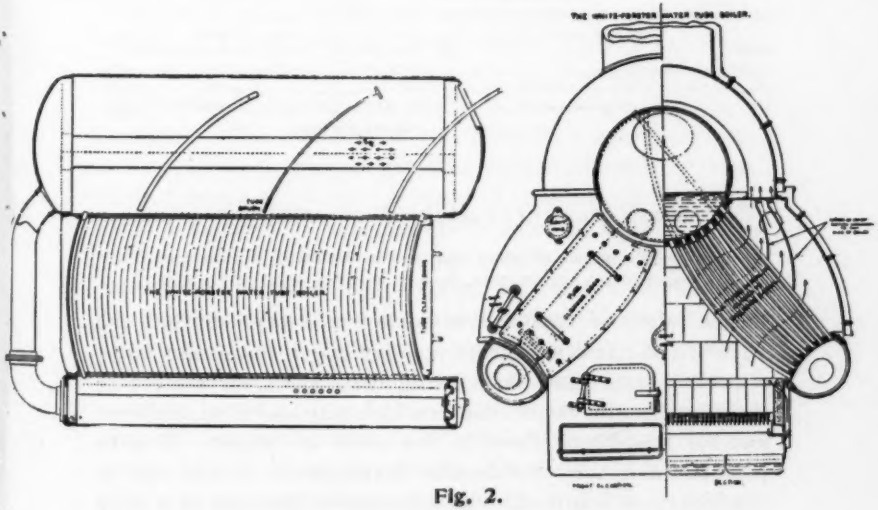


Fig. 2.

ing the air pressure. Air holes are arranged at the sides and ends for the admission of air above the grate; sight holes are fitted for the inspection of the uptakes and combustion cham-

ber. Large doors are fitted at the front end, so that the whole length of tubes is exposed for cleaning, and a brush can be passed between the tubes from end to end of the boiler.

With regard to steam efficiency, the following table shows average evaporative results from a recent test of a "White-Forster" boiler with oil fuel:

Evaporation Trial, One Boiler.	*Full Power.*	*Low Power.*
Duration of trial, hours	2	2
Heating surface, square feet	4,000	4,000
Volume of combustion chamber, cubic feet	302	302
Mean Results, per hour.		
Water evaporated (actual), pounds	45,410	15,750
Oil burnt, pounds	3,800	1,145
Water evaporated per pound of oil fuel (actual), pounds	11.94	13.75
from and at 212° F., pounds	14.5	16.7
square foot of tube surface (actual), pounds	11.35	3.93
from and at 212° F., pounds.	13.61	4.75
Oil burnt per square foot of tube surface, pounds	0.950	0.2865
combustion space, pounds	12.6	4.82
Steam pressure, pounds per square inch	205 to 212	202 to 207
Oil pressure, pounds per square inch	210 to 220	100
Air pressure in stokehold, inches of water	4.6 to 4.9	0.95 to 1.0
Oil temperature at burners, degrees Fahrenheit	200 to 215	180 to 200
Temperature of atmosphere, degrees Fahrenheit	77	80
stokehold, degrees Fahrenheit	88 max.	90 to 97
Smoke	Very light to nil.	
Number of burners in use	*11	†4
Total for two hours, Water evaporated (actual), pounds	90,820	31,500
Oil burnt, pounds	7,600	2,290

* Open ½ a turn. † Open ¼ of a turn.

VENTILATION AND REFRIGERATION OF AMMUNITION HOLDS.

Translation of abstract of paper read at the Bordeaux International Congress in Naval Architecture, June 27, 1907, by ADRIAN BOCHET.

The safety of ammunition holds has always engaged the attention of naval architects, and for many years past measures have been taken for isolating these holds, for protecting them against the various causes which may lead to an accident, and for inundating them in the event of danger. Modern powders, and the considerable development in the use of machinery on board ship, have increased the risks in a large proportion. Modern powders have excellent ballistic properties, but are also most unstable, and their instability increases very rapidly with an increase in temperature. By their grad-

ual alteration they also set free a quantity of inflammable gases which may give rise to explosive mixtures. The development in engines and boilers, in auxiliary engines, and in the extent of steam pipes laid throughout the ship, had led to a greater heating of the various compartments. The distribution of the armament, and the necessity of providing ammunition holds in proximity to the guns to be served, have often resulted in the holds being in locations that are particularly unsuitable from the point of view of temperature. Therefore, concurrently with a gradual increase in sensibility to heat, which forms one of the characteristics of modern powders, causes tending to augment the temperature of the ammunition have increased in a large proportion also, and numerous attempts have been made with a view to inure the artificial refrigeration of the holds.

Cooling by ventilation alone is a simple and safe means to prevent the accumulation of explosive gases, but is quite insufficient to ensure a decrease in the temperature of the holds when the temperature of the air outside reaches 20 degrees (say 70 degrees Fahrenheit), and when the causes making for an increase of temperature in the holds attain a certain importance. The heating of the air which enters the ship occurs rapidly on contact with the warm bulkheads, by reason of its low specific heat. A rise of 10 degrees centigrade has often been noted for courses which appeared as short and direct as possible between the upper works of the ship and the fans placed in the holds. Cooling by the means resorted to in refrigeration holds, such as those for the transport of meat, appeared at first to constitute a final solution of the problem. But this failed completely, owing to the very great difference existing between these holds and those for storing ammunition. In the former there is no ventilation, and they remain closed during the whole passage. They are also maintained at a very low temperature, and one result of this is that the small proportion of moisture contained in the local air on starting gets deposited on the products contained in the refrigerating chamber, and does not injure them in any way.

Ammunition holds, on the other hand, require ventilating. Even were one so imprudent as to do away with ventilation completely, or to reduce it in too large a measure, the requirements of the service would demand the frequent opening of the hold, thus allowing every time the outside air to penetrate into it. Finally, the temperature which it is suitable to maintain in an ammunition hold is much above 0 centigrade (32 degrees Fahrenheit). One result of this frequent renewal of the air in an ammunition hold is that the moisture it contains is condensed on all the surfaces which are maintained at a lower degree of temperature than that of the outside air, and the water which cannot be congealed, as in the case of a provision hold, trickles down the partitions of the holds and on the ammunition. This very grave disadvantage can only be avoided by cooling the air entering the ammunition hold, so that it is not at a higher temperature than that of the contents or of the walls of the hold.

The only rational means to cool ammunition holds is to ventilate them with air suitably cooled, and this deduction has received the sanction of actual practice, as it is the only method of cooling which has prevailed so far. It has been applied to fourteen French battleships, which carry together forty-three ammunition-hold cooling plants, and to eight Russian battleships, which are fitted with thirty-seven similar plants. It will be fitted also to the armored cruisers *Waldeck-Rousseau* and *Michelet*, now in course of construction. The ammunition holds of these two ships are to be provided with æro-refrigerators, in which a circulation of artificially-cooled liquid is maintained constant; the dynamo rooms are to be cooled by a similar installation, but with sea-water circulation.

Until recently the maximum temperature which was thought advisable for ammunition holds was 35 degrees centigrade (95 degrees Fahrenheit); now, however, this limit has been brought down to 30 degrees centigrade (86 degrees Fahrenheit). With the former limit direct cooling by sea-water circulation sufficed, as the temperature of the water

taken from a certain depth remained perceptibly lower than 35 degrees centigrade in all parts of the world. The latter limit, however, can only be reached by having recourse to artificial refrigeration.

The refrigerators for cooling the air consist of metallic surfaces, on one side of which the cooling liquid circulates, the air to be cooled circulating on the opposite side. A pump ensures the circulation of the liquid, and a fan that of the air. The complete apparatus are built by Mr. F. Fouché, 38 Rue des Ecluses St. Martin, Paris.

The extent to which refrigeration has to be carried depends upon the amount of heat which enters the hold; it is necessary, in the first place, to reduce to the lowest possible minimum this amount of heat. The afflux of heat in the hold is caused by radiation from the warm sides and by conductivity from the metallic pieces they contain, and which are connected to the heated sections of the ship. The methods of obtaining a thermic insulation of the hold are the following: Radiation from the warm sides can be reduced by an inside lining made of substances the conductive and emissive properties of which are low, such as cork and asbestos. Another solution is to build a double wall, to obtain an air lining, in which case it is advantageous to ensure in the double wall a circulation of air at the lowest possible temperature. The circulation of a cold liquid inside the double wall will give most satisfactory results, provided the liquid be brought sufficiently in contact with all the metallic pieces that are liable to carry heat into the hold by conductivity.

It is easy in principle to ensure the thermic insulation of the hold, but in actual practice the application of the various methods is surrounded with great difficulties, owing to the arrangement of the holds, the small space available, and the necessity of preventing every cause of damage to the ammunition. The use of simple insulating inside coverings of a thickness proportional to the heating of the walls and the available space is evidently the most easy solution of the problem. As this method has hitherto proved of sufficient

efficiency, it has alone been developed in actual practice. The use of double walls with air circulation has, however, been successfully combined with the insulation of the sides.

The use of a cold-liquid screen is surrounded by the following difficulties: In order to be efficient, there should be no break or interval in the current. It should reach the whole of the metallic pieces which are liable to cause heat in the hold by conductivity. All risks of inundating the hold in case of leakages have to be removed. The cold screen is constituted by a double wall, but it is absolutely impossible to give the space sufficient dimensions to allow a man to enter it in order to paint the plates and keep them in a good state of repair, this proscribing the use of all liquids, such as sea-water, which may corrode the plates. The solution consists in causing to circulate inside the double wall a liquid, such as fresh water charged with oil, or milk of lime, with which there is no risk of corrosion. The total quantity of liquid thus circulating has to be as low as possible, so as to prevent in any case the risk of inundation of the hold. If a leak did occur, a small quantity of liquid only would escape without risk of damaging the ammunition. Simple means suffice to show when an accident of this kind has happened.

The paper was accompanied by drawings showing how a hold could be insulated and the method followed for ventilating it with cooled air. The combination is such that the injected air cannot be hotter than the sides of the hold, which prevents all deposition of moisture. The insulation of the hold is completed by the flowing of the exhausted air round the walls. The liquid and the air for ventilation can be cooled simply by sea-water or by means of a refrigerating machine, according to the temperature required to be maintained in the hold. The amount of heat Q^1 penetrating through the walls is proportional to their surface $\int s$, to the difference of temperature between the two sides $T-t$, and to the time.

The number of heat units going through a given partition per unit of surface and time q^1, for 1 degree difference in

temperature, is determined experimentally. Numerous tests have established the figures corresponding to partitions covered with the usual insulating material. The following formula is obtained:

$$Q^1 = \int s\,(T - t)\,q^1$$

for the unit of time.

Further, the metallic parts, such as partitions, bulbs, standards, projectile supports riveted to a warm wall, cause a quantity of heat to penetrate the hold; this is

$$Q^2 = \int s \frac{q^2}{e}(T - t),$$

where s is the section of the metallic part considered; e the distance between the point where the heat maintained at a temperature T penetrates, and the point whence this heat is emitted inside the hold, the temperature of which is t; q^2 being the number of heat units passing per unit of time and per unit of section of the piece, considered between two points, with the unit of length between them, and maintained at a temperature difference of 1 degree.

Neglecting the amount of heat brought in by the *personnel*, caused by the handling of the projectiles, also the entrance of warm air, the total quantity of heat penetrating the hold will be

$$Q = Q^1 + Q^2.$$

This amount of heat has to be totally carried away by ventilation in order that the hold may be maintained at the required temperature. It is necessary, therefore, to send into the hold a volume of air V capable of absorbing the quantity of heat Q, by being heated from its entrance temperature τ to an outlet temperature θ. The calorific capacity of air being 0.3 heat unit per cubic meter, the following will be the formula:

$$V = \frac{Q}{0.3\,(\theta - \tau)}.$$

It is necessary to ascertain that the volume of air thus determined meets the ventilating conditions of the hold according to the nature of ammunition it contains.

In order to bring the volume of air V from the initial temperature θ measured at the moment it enters the refrigerating machine to the temperature θ measured at the outlet, the amount of heat given up by the air proper, by the water vapor it will contain after cooling, and that given up by the vapor which condenses, have to be deducted.

The specific heat of dry air being 0.3 heat unit (calorie) per cubic meter,

That of vapor being 0.48 heat unit (calorie) per kilogramme,

The latent heat of water evaporation being 606.5 heat units (calories) per kilogramme,

p the weight of water vapor contained in 1 cubic meter of air drawn, and

p^1 that remaining in 1 cubic meter of the air after cooling,

The formula for the heat to be absorbed will be

$$Q^1 = V[(\theta - \theta)(0.3 + 0.48\,p^1) + 606.7\,(p - p^1)].$$

In designing the refrigerating machine it is necessary to add to this amount of heat Q^1 that resulting from the heating of the apparatus themselves, and of the pipes in which cold air and liquid circulate.

The dimensions of the apparatus and of its auxiliary parts—the circulation pump and the fan—are deduced from the volume of air V flowing per unit of time, and from that of the cold liquid necessary to carry away the quantity of heat Q^1 increased as aforesaid.

It should be remarked that the power of transmission of heat of an æro-refrigerator from air to water is, so to speak, without limits; it depends solely upon the output. It increases with the speed, so that the dimensions of the apparatus, suitably proportioned, are solely dependent upon the required pressure of the air and cooling liquid.

Observations made on a number of installations show that the machines supply air easily at a temperature which does not exceed by much more than about 1 degree centigrade

that of the cooling liquid, whatever be the temperature of the air drawn into the machine. The water vapor contained in the air is completely condensed in the æro-refrigerator in a proportion which corresponds with the fall in temperature. The hold thus remains perfectly dry.

Tests made on board the *Sully*, in the Far East, have shown that with temperatures of 31 degrees centigrade (87.8 degrees Fahrenheit) for sea water, and 36 degrees centigrade (96.8 degrees Fahrenheit) for the air, the apparatus maintained the hold at a temperature of 32 to 34 degrees centigrade (89.6 to 93.2 degrees Fahrenheit).

The results of the other tests, and illustrations of the apparatus, were given in the paper.—"Engineering."

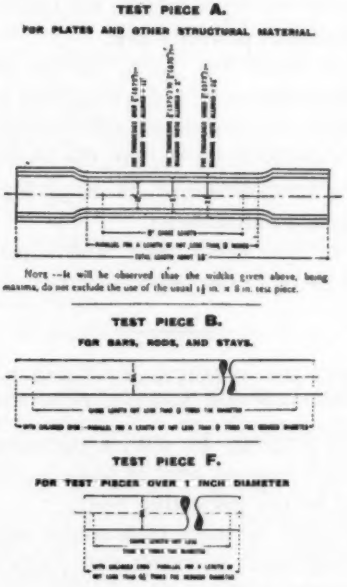
BRITISH STANDARD SPECIFICATION FOR INGOT-STEEL FORGINGS FOR MARINE PURPOSES.

Ingot steel for forgings for marine purposes may be made by the open-hearth process, acid or basic, as may be specified, and as approved by the inspecting body.

The forgings must be made from sound ingots, and for all important forgings (such as crank and propeller shafts, connecting rods, piston rods, crosshead pins, etc.) the forgings must be gradually and uniformly forged. Not more than the lower two-thirds of the ingot is to be utilized for purposes of forging. The sectional area of the body of the forging (as forged) shall not exceed one-fifth of the sectional area of the original ingot, and no part of the forging (as forged) shall have more than two-thirds of the sectional area of the original ingot.

The tensile strength of ingot-steel forgings for marine purposes, ascertained from standard test pieces, must be between the extreme limits of 28 and 40 tons per square inch. In all cases a margin of four tons per square inch shall be allowed between the specified maximum and minimum tensile breaking strengths. The elongation, measured on a standard test piece, must be not less than 29 per cent. for 28-ton steel and

17 per cent. for 40-ton steel, and in no case must the sum of the tensile breaking strength and corresponding elongation be less than 57.


TEST PIECE A.
FOR PLATES AND OTHER STRUCTURAL MATERIAL.
TEST PIECE B.
FOR BARS, RODS, AND STAYS.
TEST PIECE F.
FOR TEST PIECES OVER 1 INCH DIAMETER

STANDARD TEST PIECES.

Cold-bend tests are to be made upon test pieces having a rectangular section of 1 inch wide by $\frac{3}{4}$ inch thick. The test pieces shall be machined and the edges rounded to a radius of $\frac{1}{16}$ inch. The test pieces shall be bent over the thinner section. Bend tests may be made by pressure or by blows.

The test pieces must withstand, without fracture, being bent through an angle of 180 degrees, the internal radius of the bend being not greater than that specified below.

Maximum specified tensile strength of forging.	Internal radius of test piece after bending, *Inch.*
32 tons per square inch, ...	$\frac{1}{4}$
Above 32 and up to 36 tons per square inch,	$\frac{3}{8}$
36 and up to 40 tons per square inch,	$\frac{5}{8}$

BRITISH STANDARD STEEL BARS, ROUND AND SQUARE.

Nominal size of bar. (Diameter or width across flats.) *Inches.*	Hot rolled. Margin of manufacture. *Inch.*	Bright rolled and drawn. Margin of manufacture. *Inch.*
$\frac{1}{4}$ (.25),	+ .010	— .002
$\frac{5}{16}$ (.3125,)	+ .010	— .002
$\frac{3}{8}$ (.375),	+ .010	— .002
$\frac{7}{16}$ (.4375),	+ .010	— .002
$\frac{1}{2}$ (.5),	+ .010	— .002
$\frac{9}{16}$ (.5625),	+ .010	— .002
$\frac{5}{8}$ (.625),	+ .010	— .002
$\frac{3}{4}$ (.75),	+ .010	— .002
$\frac{7}{8}$ (.875),	+ .010	— .003
1,	+ .015	— .003
$1\frac{1}{8}$ (1.125),	+ .015	— .003
$1\frac{1}{4}$ (1.25),	+ .015	— .003
$1\frac{3}{8}$ (1.375),	+ .020	— .003
$1\frac{1}{2}$ (1.5),	+ .020	— .004
$1\frac{5}{8}$ (1.625),	+ .020	— .004
$1\frac{3}{4}$ (1.75),	+ .020	— .004
$1\frac{7}{8}$ (1.875),	+ .025	— .004
2,	+ .025	— .004
2 to 4 inches, black square,	+ .030	...
2 to 3 inches, bright round,	...	— .005
2 to 6 inches, black round,	+ .025	...

GERMAN NAVAL POLICY AND THE NORTH SEA CANAL.

A scheme necessary for the realization of the ambitious naval scheme of the German Government is the widening of the North Sea Baltic canal, which will involve a very heavy expenditure. The canal at present is of such dimensions as to limit the size, and particularly the width and draught, of battleships, and the advent of the *Dreadnought* made it imperative on the part of Germany, if her warships were to equal the new types, to practically reconstruct the canal. To

this extent the Government will be handicapped in their effort to provide the funds for the building of ships. The extent of their expenditure on works will be appreciated by a description of the canal scheme now before Parliament.

The section of the new canal, prepared in connection with the Government Bill before the German Parliament, shows a depth of 11 meters (about 36 feet) below average canal water level, and at this depth the breadth is to be 44 meters (about 144 feet); whilst the present depth is 2 meters less, and the breadth only half the proposed breadth. The breadth at the surface of the water will be about 340 feet, against 223 feet as at present, and the water-carrying section is increased from 413 square meters to 825 square meters. This profile should answer all requirements, as far as these can at present be foreseen.

These dimensions will allow of the bridges at Grünenthal and Levensau remaining, in spite of their comparatively small foundation depth. Should, at any time, a further enlargement of the canal section be required, the depth can, without difficulty, and without a very material increase in cost, be increased to 13.5 or 14 meters. Under such circumstances the two high-level bridges already referred to would, however, have to be rebuilt. In order to avoid this contingency for the new high-level bridges the foundations of the latter will be carried to a greater depth. So as not to increase too materially the cost of the extension the present line of canal will, on the whole, be adhered to.

In two places, the Obereidersee section and that between Holtenau and Levensau, a new line will be cut, so as to avoid some undesirable curves. These alterations in the direction are looked upon as indispensable. The former section (the Obereidersee) has proved unsatisfactory ever since the construction of the canal. The high banks at the curves interfere with navigation by blocking the view and the sight of approaching steamers, so that it is difficult to ascertain their speed. The section between the Levensau Bridge and the Holtenau Lock is equally awkward, inasmuch as the sharp-

ness of the curve places obstacles in the way of the passage of the large steamers, and proper regard to safe transit makes the doing away of these curves essential. The smallest permissible curve should be 1,800 meters radius, whilst the one on the Levensau-Holtenau section is only 1,200 meters. The canal line must be straight on either side of the Levensau Bridge, as the section of the bridge for large steamers is only 38 meters broad. It is also essential that vessels coming from the west should have an open view of the Holtenau inner harbor and locks. Similar considerations make it also desirable that the canal should be made distinctly broader at other sharp curves.

For the purpose of increasing the efficiency of the canal the distance between the passing places will in the future be only a little more than six miles (10 kilometers). The experience gathered from the canal traffic in the past shows that this is the shortest distance there should be, and even that is considerably more than at several other canals; at the Suez canal the distance is only about three miles. Four of the passing places will be so constructed as to allow of turning, an arrangement which will allow a fleet passing through the canal to turn in comparatively short time. For the ordinary passing places the bottom breadth is about 445 feet, and the breadth at surface of water 635 feet. At those intended for turning places the length is 3,670 feet, the breadth at bottom 550 feet, and at surface of water 730 feet, with a turning circle of 1,000 feet in diameter at the bottom.

The locks at Brunsbüttel and Holtenau will be 1,100 feet long between the gates, 150 feet broad and 46 feet deep, with ordinary canal water level—that is, Baltic water level. The locks will consequently, even with comparatively low water level, still have a depth of 40 feet. The locks of the new Kaiser Harbor at Bremen are 735 feet long, 93 feet broad and 35 feet deep at ordinary high water. The proposed depth of the canal locks will, as far as can be seen, be sufficient for the largest war and merchant ships. These dimensions are all the more important at the Brunsbüttel lock, on account of its

vicinity to the mouth of the Elbe, where damaged, deep-draught ships can be quickly conveyed to the interior harbor for repairs. In fixing the breadth due consideration has been given to the fact that the breadth is likely to increase in the shipbuilding of the future. Although the lock dimensions, perhaps, may be a little ample for the present time, the margin is looked upon as unavoidable, as a possible later enlargement of the locks would mean complete rebuilding. These large locks may also, it is claimed, prove a boon to the shipyards on the Baltic, as they will allow of the large steamers built there going through to the North Sea.—"Engineering."

BRITISH TARGET PRACTICE.

The cruiser *King Alfred*, flagship of Vice-Admiral Sir Arthur Moore, Commander-in-Chief of the China Squadron, has made the following scores while at gunnery practice at Wei-hai-wei:

With three 6-inch guns in one minute—

Rounds.	Hits.	Bulls.
11	11	11
14	13	8
13	13	9

With her two 9-inch guns in two minutes—

Rounds.	Hits.	Bulls.
10	10	8
9	9	7

Altogether eighteen big guns fired 198 rounds making 188 hits and 113 bulls. This remarkable record places the *King Alfred* far ahead of every ship in the British Fleet.

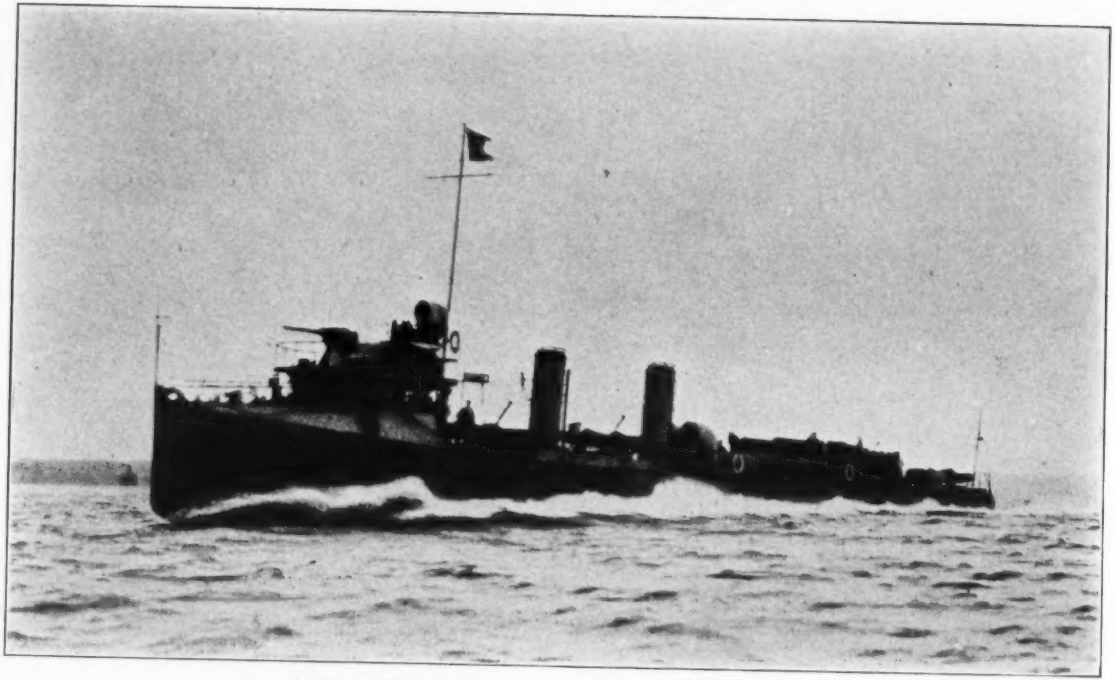


Fig. 1.—THE TORPEDO-BOAT STEAMING AT 27 KNOTS.

OIL-FUEL TURBINE-DRIVEN TORPEDO BOATS.

There are given here several illustrations of the 26-knot torpedo boats built by Messrs. J. Samuel White & Co., Ltd., of East Cowes, Isle of Wight, for the British Navy, and there is reproduced a plan of the machinery room and another view of the high-pressure turbine. These represent a new series of boats, of which twenty-four have been laid down since the first were ordered under the naval programme of 1905-6. Up to that time the torpedo boats had ranged in speed up to 25 knots, the largest being of 205 tons displacement, with a length of 165 feet, 17 feet 6 inches beam, and 5 feet 6 inches draught; the horsepower of the reciprocating machinery being 3,000. Those vessels were completed three or four years ago. Later it was considered that a superior type of boat was necessary for even the tactical work undertaken by purely torpedo craft, in contradistinction to the over-sea offensive work of destroyers. The Board of Admiralty therefore decided to introduce a new type, which were at first classed as "coastal torpedo-boat destroyers," but this has since been changed to torpedo boats. In view of the great suitability of the Parsons turbine, as ascertained in exhaustive trials with successive destroyers—notably the *Eden*—this type of propulsive machinery was adopted. At the same time the Committee appointed over to investigate the advantages of oil fuel had solved the difficulties of the system, not only as regards the best means of vaporizing the fuel, but also of completely consuming it, and the new boats were therefore designed with boilers to use only liquid fuel. As we have said, twelve boats were ordered, the design being left to the three firms with whom the contracts were placed.

Messrs. J. Samuel White & Co., Limited, of East Cowes, secured the order for five, which was not surprising to those cognizant of the work done by the Isle of Wight firm, not only in connection with torpedo craft, but with high-speed light-draught vessels. The steady development of the torpedo boat, from the advent of the *Lightning*, about thirty years ago, had forced upon the minds of tacticians the neces-

sity for a vessel of the same type and of greater power to act as catcher or destroyer, and Messrs. J. Samuel White & Co., of Cowes, were the first to produce such a catcher, one vessel —the *Swift*—passing through successful trials in 1885, while another, two years later, of the same type but of larger dimensions, proved equally successful, and was purchased by the then firm of Sir W. G. Armstrong, Mitchell & Co., and subsequently added to the Chinese Navy. Since then the firm have kept pace with the requirements of the naval strategist, and proof of their success is found in the fact that out of thirty-two of the modern torpedo craft with oil fuel and Parsons turbines, ranging from 26 to 36 knots speed, now under order, eleven are being built by the Isle of Wight firm.

When the order for the vessel illustrated was placed, the firm were in an advantageous position for carrying out the undertaking. They held a license from Messrs. Parsons for turbine construction, but although no such engines had been made, new machine tools were rapidly installed for the purpose. This, however, was a matter of comparative simplicity. What was of more consequence was the necessity of securing the minimum scantlings for the hulls with absolute oil-tightness in those parts in which the fuel was to be stored. A complete series of experiments were entered upon, culminating in the construction of a short length of vessel, of the full section, in order to practically test the system of riveting proposed. This enabled the constructional details to be settled, and it is interesting to note that other firms followed the same lines, and a paper on the subject was read at the Institution of Naval Architects last year dealing with experiments of the same description as those previously carried out at Messrs. White's works (see "Engineering," vol. lxxxi, pages 511 and 802). The details of scantling thus having been determined, rapid progress was made with the construction of the first five vessels. All the boats were delivered before the contract date, the first anticipating the time by a month, and the fifth, which was handed over on the 8th of May last, by nearly three months.

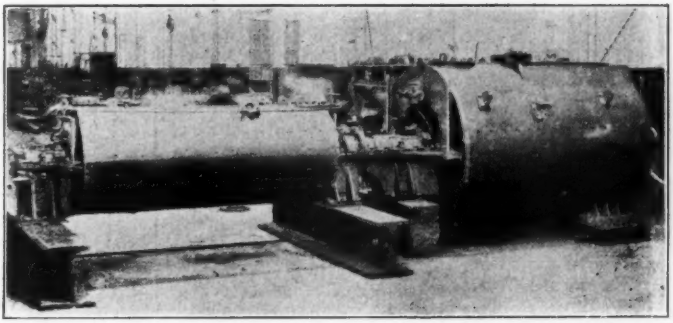


Fig. 2.—The Cruising Low Pressure, and Astern Turbines.

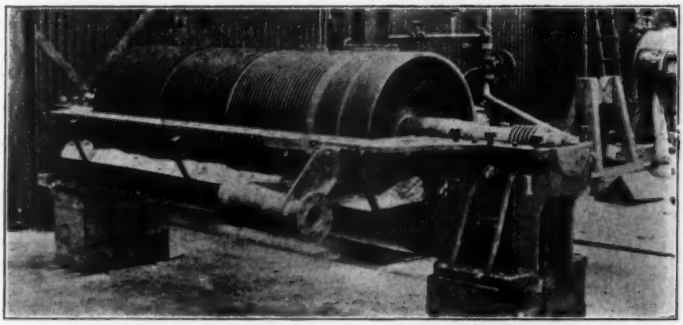


Fig. 3.—The Cruising Turbine with Top Cover Off.

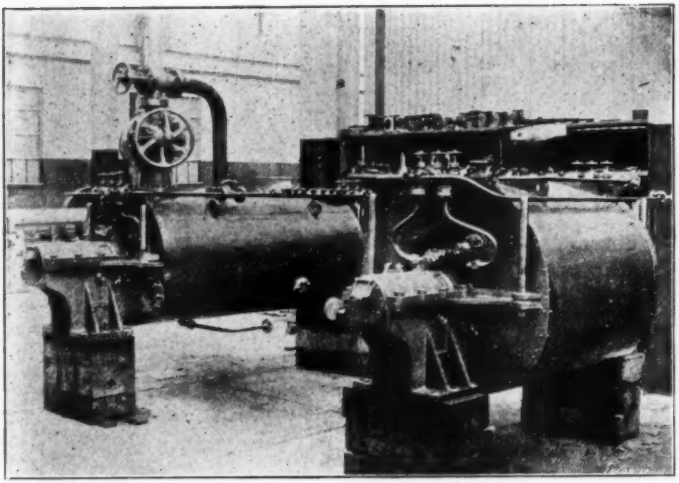


Fig. 4.—The High-Pressure and Intermediate-Pressure Ahead Turbines.

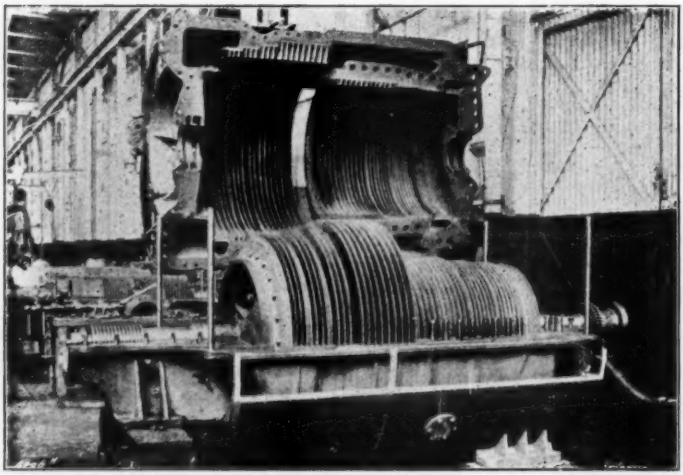


Fig. 5.—The Low-Pressure Ahead and the Astern Turbine (Combined).

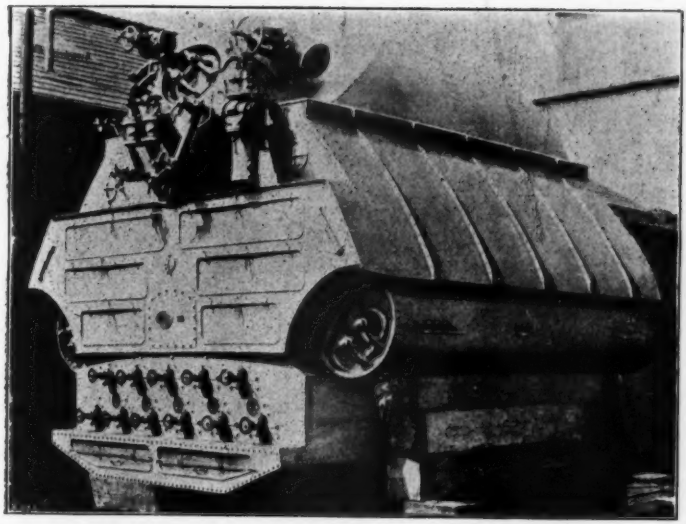


Fig. 6.—The White-Forster Water-Tube Boiler.

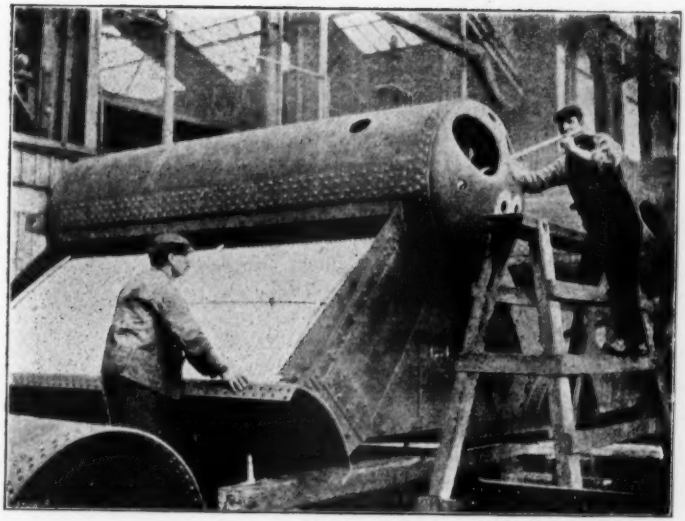


Fig. 7.—Method of Tubing White-Forster Boilers.

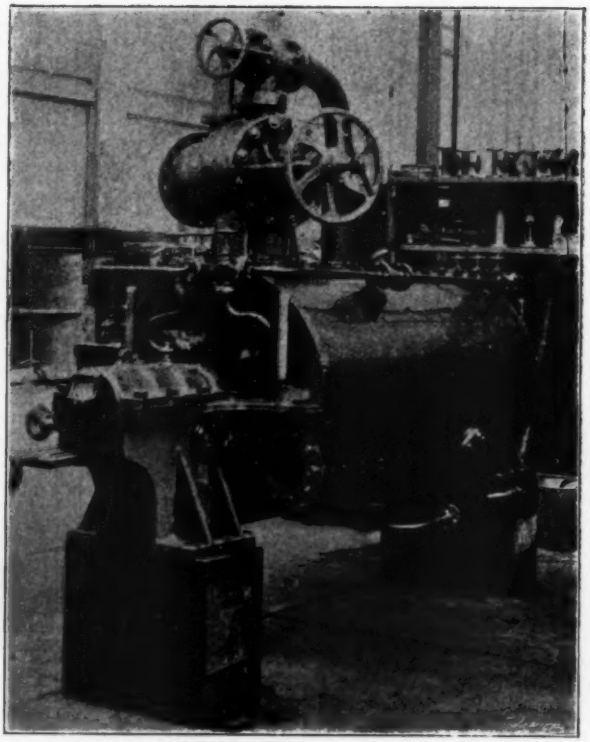


Fig. 9.—HIGH-PRESSURE AHEAD TURBINE.

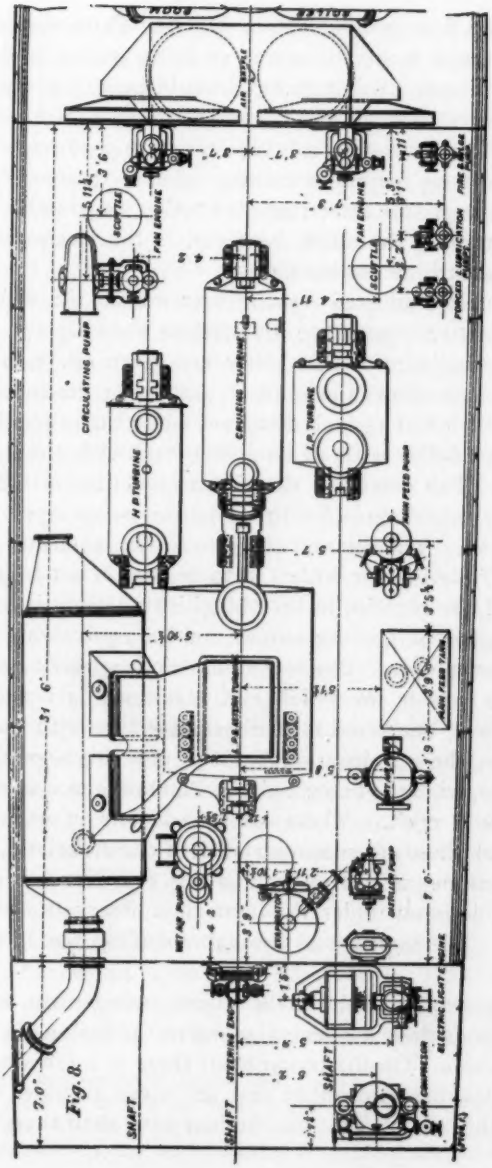

CIRCULATING PUMP
H.P. TURBINE
CRUISING TURBINE
L.P. TURBINE
MAIN FEED PUMP
AUX. FEED TURB.
SHAFT
SHAFT
SHAFT

Fig. 8.

The first five vessels built by Messrs. White were 175 feet long, 17 feet 6 inches beam and 10 feet 9 inches in depth, the displacement at 5 feet 8 inches draught being 245 tons. The vessels, as shown in Fig. 1, are fitted with a high whale-backed forecastle, with a bridge carrying a 12-pounder gun, while the three torpedo-launching tubes are mounted at intervals on the middle line; there is another 12-pounder gun aft. The feature of the ships, however, is the steam-generating plant and turbine machinery.

The boilers adopted by the firm are of the well-known White-Forster type. Two illustrations showing the distinctive characteristics of this boiler are given (Figs. 6 and 7), one of which shows the work of putting in the tubes, which are of 1-inch and 1⅛-inch diameter, while the other shows the completed boiler with all its fittings and with the liquid-fuel nozzles. The details of the oil-fuel installation—the result of costly experiments by the Admiralty—are very properly kept secret. The relation of grate surface to heating surface in the White-Forster boilers in these vessels is 1 to 60 square feet, and the practice in liquid-fuel installations is to provide 2 square feet of heating surface for the equivalent of 1 indicated horsepower. The boilers have proved very satisfactory in these vessels, to which fact testimony is borne by the boiler being bracketed as an alternative type with the Yarrow boiler in the specification for the new high-speed cruiser *Boadicea*, which is being built at Pembroke as a mother boat for torpedo craft. There are two boilers in each torpedo boat, and these are accommodated in one stokehold, the fans for which are, as will be seen in Fig. 8, driven by engines in the engine room under the immediate observation of the engineer. The shaft passes through a stuffing box in the bulkhead.

The general plan shows the engine arrangement, and on the other plates there are several views of the turbine in course of construction. On the center shaft there is a cruising turbine and a low-pressure ahead and an astern turbine, the latter two within one casing. On the port wing shaft there is a high-

pressure ahead turbine, and on the starboard side an intermediate-pressure ahead turbine. All four ahead turbines are in use for speeds up to 16 knots, steam from the boiler being passed to the cruising turbine, thence to the high-pressure turbine on the port shaft, to the intermediate-pressure turbine on the starboard shaft, to the low-pressure turbine on the center shaft, and, finally, to the condenser, which is placed on the port side of the ship. This arrangement of four compound turbines gives a great range of expansion, and has resulted in satisfactory economy even at low powers. For speeds above 16 knots the steam is shut off from the cruising turbine—which then works *in vacuo*—and passes direct to the high-pressure turbine, thence to the intermediate, the low-pressure, and finally to the condenser. The turbines, which were constructed by Messrs. J. Samuel White & Co., are bladed on the Parsons system, and the details correspond with those generally adopted in torpedo craft. Fig. 2 shows the cruising turbine, with the low-pressure ahead and astern turbines to the rear. Fig. 3 illustrates the rotor of the cruising turbine, from which it will be seen that there are three stages. Fig. 4 shows the high-pressure and intermediate turbines, which are considerably shorter than the cruising turbine. On this view there is seen the valve for the passage of steam from the boiler direct to the high-pressure turbine. Fig. 5 is specially interesting, as it shows the rotor of the low-pressure and astern turbines in place, with the upper part of the casing raised, and the thrust shaft and vertical supports for lifting the upper part of the casing. Fig. 9 illustrates the high-pressure turbine complete, with its steam valves.

The contract called for the maintenance of a speed of 26 knots during an eight-hours' run, with 21 tons of oil fuel on board, and all five vessels easily maintained this condition. The mean results of the five boats were as follows:

Speed on measured mile (Stokes Bay), knots	27.107
Mean speed for eight hours, knots	26.36
Consumption of oil for eight hours, tons	20.758
Distance run per ton of fuel at a speed of 12.147 knots, nautical miles	37.90

It should be noted that as the depth of water at Stokes Bay is not quite adequate for such high speeds the vessel will, in deep water, exceed the 27.107 knots attained on the measured mile. The consumption of oil averaged just under 2.6 tons per hour for 26.36 knots. On the earlier torpedo boats of 25 knots speed, of which Messrs. J. Samuel White & Co. built several, the consumption was also 2.6 tons per hour, but the speed was lower (1 mile per hour), and the displacement was 30 tons less than the new oil-fuel turbine-driven boats, so that the result must be pronounced very satisfactory. It will be noted also that at 12 knots the vessels steamed 37.99 nautical miles for each ton of fuel consumed, whereas the guarantee under the contract was 32 nautical miles, so that here, as in the matter of speed, there was a considerable margin over the stipulated condition.

Messrs. J. S. White & Co. are building four more vessels of this type; the dimensions have been slightly increased—the length by 7 feet, the beam by 6 inches and the displacement by 10 tons—so that the new vessels are 182 feet long, 18 feet beam, and at 5 feet 10 inches draught they will displace 256 tons. The estimated power to give them 26 knots is 4,000 horsepower. Two 33-knot destroyers are also in course of construction. These are 272 feet in length, have a beam of 26 feet, and at 8 feet 8 inches draught the displacement is expected to be about 900 tons. It is anticipated that the 33 knots will be realized with a power equivalent to 15,500 indicated horsepower. Five other vessels of this class are being built, in addition to a 36-knot destroyer by other firms, and the performances of these various vessels on trial will be looked forward to with considerable interest.—"Engineering."

RECIPROCATING ENGINES FOR OCEAN-GOING STEAMERS.

Paper read at the Engineering Conference of the Institution of Civil Engineers, Section III, Machinery, June 19, 1907, by HENRY DAVEY, M. Inst. C. E.

The propelling power of the ship begins with the fuel and ends with the screw. The real fuel economy is the fuel required to drive the ship at a given speed. If the turbine requires a screw which is necessarily less efficient than that of the piston engine, it is the fault of the turbine system. Leaving out the boiler efficiency, the propeller of the ship is the engine and screw combined. In this connection, what is wanted for the purpose of ascertaining detail losses is not only the actual mechanical efficiency of the turbine, which Mr. Denny is trying to obtain by measuring the torsion of the shaft, but the actual thrust exerted by the screw in propelling the ship. I think an indicator for that purpose might be devised, but it should be an indicator for use with the ship in motion under normal conditions.

The subject of ordinary reciprocating, or, to use a shorter word, piston, engines can better be discussed by comparison with rival systems, such as turbine and gas or oil engines, although the two latter are not at present used for large ocean-going steamers.

On the question of fuel economy alone I have not been able to obtain any reliable information which would enable me to compare the piston engine with the turbine; but it does appear that the turbine has not beaten the piston engine in over-all efficiency—that is, taking engine and screw together.

Economy of fuel must always be one of the most important factors in steamship working, and it may be that the competition of the turbine will lead to improvement in the fuel economy of the piston engine, and possibly in an improved system of working. The turbine will doubtless be improved also.

There is little hope of improved fuel economy from increased boiler pressure alone. The theoretical possibilities of

saturated steam at 100 pounds, 150 pounds and 200 pounds, and at 27.5 inches vacuum, are $8\frac{3}{4}$ pounds, 8 pounds and $7\frac{1}{2}$ pounds of steam per indicated horsepower respectively. The best land engines give an efficiency ratio of about 70 per cent., corresponding with 12.5 pounds, 11.4 pounds, and 10.6 pounds respectively. That is a much better economy than is usually obtained with high-class marine engines.

High-class land engines, owing to the use of the Corliss and drop types of steam valves, have much less clearance space; they have also a much better steam distribution, and often have reheaters between the cylinders. I do not see why Corliss or drop valves should not be used in marine engines. They might be placed in the covers of the cylinders, and the clearance spaces would in that way be greatly reduced. Reheating between the cylinders might also be adopted on a plan to be named further on.

The introduction of the turbine has given rise to a new arrangement of engines, an arrangement which appears to me to be advantageous for piston engines also—namely, the arrangement of three screw shafts with a turbine on each shaft, the center turbine, or high-pressure one, exhausting into, and supplying the steam for, the two low-pressure turbines working the side, or wing, screws. This arrangement of engines and distribution of steam appears to me to be suitable also for piston engines. Each piston engine would have a separate reversing gear, and the gears would be actuated separately or together as desired. Another arrangement would be that of having a high-pressure piston engine for the center screw, exhausting into low-pressure turbines working the wing screws. It would occupy too much space to go into details, but the scheme is worth consideration, as it would lend itself to a system of superheating the steam between the high and low-pressure engines. In many land engines steam is superheated between the cylinders by means of reheating coils supplied with boiler-pressure steam, but the amount of heat thus added to the steam is not sufficient to effect much saving. What is wanted is an effective superheater for the

purpose. It does not appear practicable to supply the heat from the boiler, but a separately-fired superheater, fired with gas from a gas producer, does not appear to present the same difficulties. An ordinary gas producer without gas-cleaning plant would answer the purpose. If a practicable system of what I will call stage superheating could be worked out, greatly increased economy would result from its application to both piston and turbine engines.

The piston engine appears to be able to compete favorably at present with the turbine in every respect, except that with the turbine there is not the thud of the piston engine, there is probably much less vibration, and there appears to be a considerable reduction in the engine-room staff. Also, the cost of upkeep may be less; but only time and extended experience can satisfactorily determine these questions. I do not know that the vibration due to screws alone would be more with the piston engine and three screws than it is with the turbine and three screws. Probably, as the cost of fuel is so important an item in the total cost, the race between the turbine and the piston engine will be won on the coal bill.

Use of Gas Engines on Board Ship.—I am of opinion that there is little hope of the marine gas engine proving more economical in fuel than the piston or turbine engine. Apart from the question of fuel, the mechanical difficulties connected with the use of large gas engines for such a purpose are very formidable. If bituminous coal is used, a very extensive gas-cleaning plant will be required, a plant quite unsuitable for use in a ship. If the gas is not thoroughly cleansed, tar gets into the engine and gives rise to troubles which could not be put up with during a voyage. The consumption of coal would probably be about 1¾ pounds per brake horsepower, a result which might be easily beaten by the present or improved piston engines or turbines.

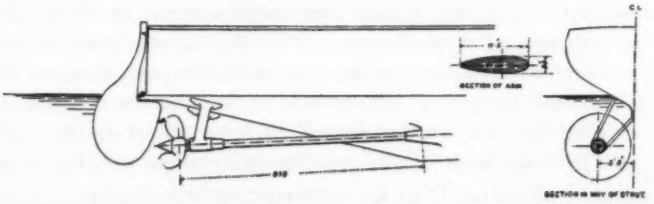
A better use of gas on board ship would be for the purpose of stage steam superheating for piston or turbine engines. A gas-cleaning plant would not be required in that application. Assuming the piston or turbine engine to be made

capable of working with an economy of 10 pounds of steam per indicated horsepower hour, the result would be much better than that likely to be obtained from the gas engine, even if the engine could be practically applied and made satisfactory in other respects.—"Engineering."

PROPELLER STRUTS.

Read before the Institution of Naval Architects, March 21, 1907, by GEORGE SIMPSON.

The subject of suitable area for propeller-strut arms is one which, so far as the writer is aware, has not hitherto been dealt with in the transactions of this or kindred institutions, and has, for some reason or other not explained, been omitted from the rules of the classification societies to whose requirements so great a proportion of our vessels are built. With



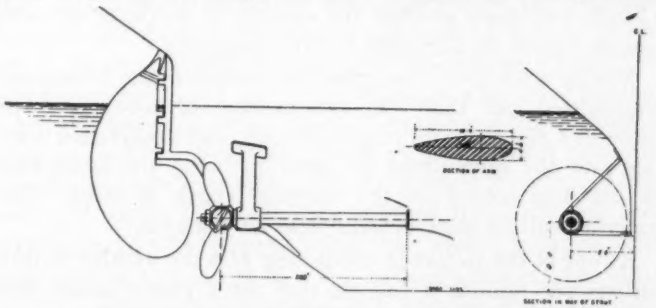
STRUTS FOR THE PROPELLER SHAFTS ON A TORPEDO BOAT.

the introduction of turbines for marine propulsion, and the consequent increase in the number of shafts, coupled with the fact that these structural fittings are of at least equal importance to rudder stocks, whose diameters have always been the subject of specific requirements, the subject of proportional sizes for these struts becomes worthy of interest. The object of the paper is to induce discussion on a semi-empirical formula, whose function is the determination of suitable dimensions for strut arms by a definite method, with a view to insuring proportional relationship between the area of arm, the horsepower transmitted and the overhang of struts.

The present almost general practice of proportioning these directly on the horsepower and previous experience, or in

many cases individual opinion, has resulted in widely divergent results for similar work. This is probably more particularly noticeable in the size of strut arms for the larger classes of naval vessels, where in many cases we find about twice the sectional area which would be assigned for an ocean liner of similar power, revolutions and overhang, although structurally the naval vessels are of lighter construction. The result is that many tons of valuable material and weight are lost to the builder and owner respectively.

It would seem almost impossible to determine the stresses exerted on these strut arms, although the impression at first sight tends to the view that the pressure exerted on the boss journal by the overhang of arm would give a moment suffi-



STRUTS FOR THE PROPELLER SHAFTS ON A TWIN-SCREW FREIGHT STEAMER.

ciently accurate to calculate an equivalent resisting moment of section with a suitable fiber stress. This view of the case, considering the bracket as a cantilever, would result in giving the greatest section modulus to the arms where they adjoin the shell palm, and tapering from thence to the junction with the hub. This was not an unusual design of struts when twin-screw vessels first came into general use. The few cases, however, in which propeller struts were carried away, showed conclusively that the greatest stresses, apart altogether from accidental causes, were those induced by centrifugal forces, and were alternating ones, causing vibration to the outer part of the arms where they joined the hub. These stresses should

be very slight with a perfectly-balanced propeller, and would be greatly augmented by a badly-balanced wheel or in a vessel of light construction, where there is a tendency for the after part of the vessel to "wiggle." That such a movement does take place to a very appreciable amount came under the immediate notice of the writer on one of the 27-knot destroyer class having a rudder of the usual overhung type, and a stock of exceptionally large diameter and of 36-ton steel, where the distance between the side of rudder and the propeller tips was 1½ inches measured in dock, and after trial trip, on redocking, the propeller tips were found bent over and the rudder plates excoriated with contact. The deflection of the rudder as calculated accounted for only about ⅜ inch of the original 1½-inch clearance, so that the contact of propeller and ship was in a great measure due to this "wiggling."

The formula given comprises the factors P, the horsepower transmitted per shaft; R, the number of revolutions per minute; l, the outboard length of shaft, and a coefficient varying with the revolutions, its function being the adjustment of the area solved to the varying length of strut. This function will be more evident from an example.

As one of the factors in computing P is the number of revolutions per minute, it follows that for a given power there may be a great difference in the revolutions per minute, and consequently in the diameter of propeller, the larger wheel requiring a much longer strut to hold the propeller clear of deadwood, and, on the other hand, a high-turning propeller would take a comparatively short strut. It will be evident from this that, viewed as a cantilever, the shorter strut, having the lesser bending moment, would not require so great a section modulus to produce the equivalent resisting moment. In the example stated we have taken the case of $P = 2{,}000$, transmitted through a shaft turning at 80 revolutions per minute in a freighter, and a similar power given in a torpedo-boat shaft turning at 400 revolutions per minute. The outboard length l in the first case is taken at 300 inches and at 310 inches in the high-speed vessel, as the latter would have

much finer lines, accounting for a greater distance for the intersection of the center line of shaft with the side of vessel than would be the case in the fuller-lined freighter. We then have—

		For the freighter. *Inches.*	For the torpedo boat. *Inches.*
Area	$= \sqrt[3]{\frac{P \cdot R \cdot l}{.0633 K}}$	= 73.0	24.9
Length of section L	$= \sqrt{5.333 \text{Area}}$	= 19.7	11.5
Breadth of section B	$= \frac{L}{4}$	= 4.9	2.8

If we assume 15 feet as the diameter of the freighter's propeller, and 6 feet 9 inches for the torpedo boat, the diagram produced will show the difference in length of strut. If we were to assume a weight for propeller and outboard length of shaft for each case and apply this weight as a deadload for the purpose of comparing the *relative* intensity of the moments acting on each, we should find a very close agreement between the ratios of these compared with the moduli of section of the arms. Of course, it must not be inferred from this that the writer claims for the formula other than what it is, viz : a rule based on observation and practice, with an attempt, as already stated, to secure a fairer relationship between the scantlings of struts and the work they are designed for.

The vessel referred to earlier in this article afterwards broke both of her propeller struts, with an interval of about three months between the accidents. Of course, it may not be right to attribute this altogether to the wiggling movement of the stern previously mentioned ; but there can be no doubt that it may have been a factor of no small amount in causing the fracture. The original struts of this destroyer were tapered from 7 by 1½ inches to 9 by 1½ inches with a minimum sectional area, where they broke, of 8 square inches. It will be seen on reference to ship No. 28 in the table of dimensions, that by the formula given, these struts should have been 29.8 square inches, and of section 12.6 by 3.1 inches.

The other cases in which there is a great difference between actual struts and the formula are Nos. 4 and 30, and in each of these vessels there is no doubt that the struts are dangerously near the breaking point, the extreme frailty of the arms being quite observable on the ship.

DIMENSIONS OF PROPELLER STRUTS AS FITTED AND BY FORMULA.

No.	Per shaft.				As fitted.		Per formula.	
	P.	R.	l, inches.	$\sqrt[3]{PRl}$	$L \times B$, inches.	Area, square inches.	$L \times B$, inches.	Area, square inches.
1 C	13,000	126	648	1,020	33 × 10	259	26.2 × 6.5	128.0
2 C	13,000	123	552	959	27 × 7	148.2	25.6 × 6.4	123.0
3 C	10,500	138	668	1,037	28 × 8	172.8	25.0 × 6.2	120.0
4 S	8,000	200	289	773	20 × 3	45.0	18.1 × 4.5	61.4
5 C	8,000	154	642	925	24 × 6½	122.2	22.5 × 5.6	94.8
6 C	2,250	172	372	524	16 × 4	48.0	16.0 × 4.0	48.0
7 B	9,500	120	319	714	30 × 8	180.0	22.6 × 5.6	95.5
8 B	8,700	125.5	296	686	24 × 6½	122.2	21.4 × 5.4	86.4
9 B	8,250	120	357	714	33 × 10	250.0	22.6 × 5.6	95.5
10 B	7,600	128	321	678	24 × 6½	122.2	21.2 × 5.3	83.7
11 B	5,000	120	356	597	24 × 9	169.5	20.5 × 5.1	78.5
12 L	5,350	90	390	573	24 × 6½	122.2	23.2 × 5.8	100.6
13 CS	3,000	150	350	540	16 × 4½	54.0	17.4 × 4.3	56.8
14 CS	3,000	192	304	560	18 × 4	56.5	15.7 × 3.9	46.1
15 PC	4,700	111	360	573	18 × 4½	63.6	20.8 × 5.2	81.5
16 PC	4,600	105	340	548	18 × 5	70.6	20.9 × 5.2	82.4
17 PC	4,350	113	340	551	18 × 5	70.6	20.3 × 5.1	77.0
18 PC	4,000	235	528	792	20 × 6	90.0	19.3 × 4.8	70.0
19 PC	3,670	106	360	520	18 × 4½	63.6	20.3 × 5 1	77.5
20 PC	3,000	112	368	498	18 × 4	56.5	19 3 × 4.8	70.2
21 PC	2,600	130	290	461	18 × 4	56.5	17.3 × 4.3	56.0
22 PC	1,500	100	169	294	9 × 3½	24.7	15.6 × 3.9	46.4
23 PC	1,300	125	304	367	15 × 3½	41.2	15.6 × 3.9	46.4
24 PC	840	156	408	377	12 × 4	37.5	14.3 × 3.6	38.2
25 F	3,500	125	630	651	18 × 4½	63.6	20.9 × 5.2	82.25
26 F	2,810	125	337	491	22 × 3½	65.0	18.2 × 4.5	62.0
27 D	4,200	364	483	904	15½ × 2½	27.0	14.5 × 3.6	39.3
28 D	2,420	370	380	698	8 × 1½	12.0	12.6 × 3.1	29.8
29 T*	2,000	375	250	572	7 × 1½	10.0	11.3 × 2.8	24.1
30 M†	165	180	110	148	5 × 1½	7.0	8.3 × 2.0	13.0
31 R	150	400	100	181	5 × 1½	5.5	6.2 × 1.5	7.1
32 Y	200	150	180	176	8 × 2	13.0	9.9 × 2.5	18.5
33 P*	300	600	90	253	5 × 1½	4.5	5.9 × 1.5	6.66
34 P*	250	500	72	208	5 × 1	4.0	5.9 × 1.5	6.6
35 P*	200	550	80	207	3½ × ¾	2.2	5.6 × 1.4	5.9
36 P*	93	500	72	149	3½ × ⅝	1.8	5.0 × 1.25	4.7
37 P*	17	450	48	71	2½ × 11/16	1.5	3.6 × 0.9	2.5
38 P*	10	400	42	55	2 × ⅝	0.6	3.4 × 0.85	2.2

*Single screw with vertical strut. †Strut was perceptibly too light.

C = cruiser. S = scout. B = battleship. L = Liner. CS = channel steamer. PC = passenger and cargo steamer. F = freighter. D = destroyer. T = torpedo boat. M = miner. R = river launch. Y = yacht. P = pinnace.

In support of the statement that in many cases the area of propeller-strut arms is assumed without due consideration, attention is directed to Nos. 1 and 2 in the table of dimensions, where it is shown that in almost similar cruisers a difference amounting to 75 per cent. is observed between the two vessels, showing that, as the lighter strut has performed its work satisfactorily on service, a weight of about 23,000 pounds of cast steel has been sacrificed by the builder, and its equivalent lost in ammunition space. This is not an isolated case, and it is borne out by many examples.

An interesting point to be noted in comparing the arms of struts required for single-screw vessels, where these are fitted vertically, is the much smaller area which the actual struts have in comparison with the formula results. With horizontal struts the case is reversed—that is, in the majority of cases the formula gives a slightly increased area over the actual brackets, except in the case of battleships and cruisers. This is probably accounted for by the fact that in the vertical position, the strut is theoretically not subjected to stresses as great as when it is arranged horizontally.

It would be extremely difficult to state theoretically what fastening is required through the palms to the shell; but an investigation of a great number of struts showed that the proportion of sectional area of fastenings to area of strut arm ranged from 0.4 to 0.55, and the writer is of the opinion that 0.5 would be a very good proportion for rivet or bolt area, and one that should give an ample margin of strength, as the stresses which these have to bear do not appear to be excessive.

In conclusion, it may be stated that the author has applied this formula for the past eight years to the determination of the propeller struts of a great number of vessels, in which the factors varied to the extremes found in such vessels as pinnaces, destroyers, scouts, yachts, freighters and liners. The table given will show how the results in these cases would have compared with everyday practice, had the scantlings been assumed in the ordinary way, the probability being that

in applying the formula a considerable saving in weight would have been effected.—" International Marine Engineering."

NEW VIEWS OF THE CAUSE OF THE CORROSION OF IRON.

At the recent meeting of the American Society for Testing Materials, Dr. Allerton S. Cushman, of the United States Department of Agriculture, made the first public announcement of the very interesting investigations he has been carrying on for several years on the causes which underlie the corrosion of iron. A number of new points were brought out, among which the most startling are that oxygen plays only a secondary rôle in the rusting of iron and that the best preventatives of rust are to be found among the most effective oxidizing agents known, such as chromic acid and its salts. This view is so contrary to all previous conceptions that it is naturally received with some incredulity when first heard, yet those who are familiar with the investigations and conceptions upon which the new theory of corrosion is based are of the opinion that the evidence which has been brought forward is not only convincing but conclusive. Dr. Charles B. Dudley, whose conservatism in expressing opinions on theoretical subjects is well known, stated at the close of the presentation of Dr. Cushman's paper that he considered this work to be the most important contribution to a metallurgical problem that has been made in the last quarter century. The same opinion was expressed privately by others who are familiar with the work.

The fact that chromic acid and its salts act as inhibitors of rusting has been known for some time, but no explanation of the curious phenomenon has ever been offered heretofore nor has its application to practice ever been suggested. Dr. Cushman has made a special study of this problem, and although it remains to be seen what practical benefits may develop out of these new ideas, it is most gratifying to be able to state that if any patents are granted covering rust inhib-

itors they will be taken out in accordance with the practice of the Department of Agriculture, so that they will be free for all American citizens. The investigation was begun by Dr. Cushman as a part of his study of the corrosion of fence wire, in the interest of the farmer. A farmers' bulletin on this subject was published by the Department of Agriculture about two years ago. It is safe to say that the public benefit which is likely to follow these investigations is on a par with the important economical benefits which will probably be derived from the same author's researches on the decomposition of feldspathic rocks and their use as fertilizers. Both of these investigations, made in the interests of the people, demonstrate the great national benefit which is derived from the scientific research of our federal departments, conducted by specialists of the highest rank. It is to be hoped that Congress will not fail to appreciate the advisability if not the necessity for providing the funds to carry on work of this kind, the expenditure for which is so small in relation to the public benefit.

If a text-book is consulted for an explanation of the rusting of iron it will be found that carbonic acid has heretofore been generally held responsible for the formation of rust. Iron is supposed to be attacked by carbonic acid, with the formation of carbonate, which is then acted on by water and the oxygen of the air to form the red hydroxide known as rust, the carbonic acid being again set free to take up its destructive work. According to this theory, in an atmosphere which did not, like that of this earth, contain about four one-hundredths of one per cent. of carbonic acid, the rusting of iron would be an unknown phenomenon. That this, as well as the peroxide hypothesis which has lately been developed in England, must be relegated to the dump pile of abandoned theories, seems to be conclusively shown by these latest researches.

According to the electro-chemical or electrolytic theory which Dr. Cushman upholds, the first attack on iron is not made by oxygen, even in the presence of water, but by hydrogen in the form of the hydrogen ion. According to the mod-

ern theory of solutions, many substances when dissolved in water are dissociated into ions, which may be regarded as atoms carrying static electrical charges. Water itself, even when pure, contains a certain proportion of hydrogen ions, and the presence of many impurities, especially those which are by nature acid, increases the hydrogen ions and thus the tendency to attack iron and carry on corrosion. The action is entirely electrolytic, being continually accompanied by an exchange of the electro-static relations between the iron and the attacking hydrogen. Such oxidizing agents as the chromate and bichromate of potash inhibit rusting by polarizing the iron to the condition of an oxygen electrode, thus preventing the approach or attack of the hydrogen ion. One of the most extraordinary points brought out is that this polarization effect is to some extent lasting. That is to say, if iron is immersed or "pickled" in a concentrated solution of bichromate acid and is then washed and wiped, it is rendered passive, so that it resists electro-chemical attack, whether this take the form of rust formation or the well-known plating out of copper which takes place if the chromated specimen is immersed in a dilute solution of copper sulphate. In short, the action which goes on when iron rusts is in every respect analagous to that which takes place when iron is immersed in a solution of a copper salt. In the later case, copper ions carrying positive electrostatic charges are present, iron passes into solution and assumes the electro-static charge, while copper plates out and becomes visible. When iron rusts, iron passes into solution while hydrogen "plates out." Once in solution the oxygen of the air oxidizes the iron to the insoluble form of the red hydroxide known as rust. This electrolytic action can be shown taking place by the use of a special polar indicator which has been called "ferroxyl." It follows from this that anything that will inhibit electrolytic action will act the part of a rust preventative.

To what extent the various salts of chromic acid will come into use for the treatment of boiler-feed waters and for "pickling" structural material will depend upon experiments car-

ried out on a large scale. Dr. Cushman himself is emphatic in pointing out the necessity for care and conservativeness in approaching the practical application of these purely scientific investigations. One of the modern problems in boiler practice is the rapid corrosion of boiler tubes used in connection with turbine engines. The copper which is dissolved by the action of the steam jets impinging on the bronze blades of the turbine rapidly corrodes the iron in the boilers by the electrolytic action just described. Since it has been found that the presence of bichromates in feed water will prevent this action, it seems as though the solution of this important problem has been discovered. The engineering world will eagerly await the detailed publication of Dr. Cushman's researches, as well as the results of the practical tests which are sure to follow.—"The Engineering Record."

GLAND TROUBLES WITH A STEAM TURBINE.

By Edward Russell.

The old type of steam-sealed gland is used in the majority of turbines, running or under construction, in England, whereas in America the water-seal type is being adopted in all cases. The only complaint I have heard against the latter gland came from Kimberley, South Africa, where the water is so bad that it used to fill the water space in the gland with solid deposit, but this has been since overcome by using condensed water.

The water gland appears to be by far the more economical. For instance, with a machine running on a traction load, the exhaust from the relay valve cannot be used, for the simple reason that this varies exactly with the load, giving the largest exhaust when the load is the lightest. This would be a very serious drawback in a traction station, because at the heaviest load the glands would receive very little steam, with resultant loss of two or three inches of vacuum. This is overcome in traction work by sealing the glands with live steam, which is very wasteful practice on account of the relay

exhaust being entirely lost. In a large turbine this exhaust will amount to a number of pounds in an hour. Of course, in throttle governing, such as used by Willans & Robinson, this is eliminated.

In lighting stations where the load is practically steady the relay exhaust works very well, but even here it has to be assisted with a little live steam. Vacuum has been seen to vary between three and four inches with steam from the relay valve alone; then when the load has practically dropped to zero, the steam has blown out from each gland in sufficient quantity to fill the engine room.

Then, too, I have noticed another thing. The old-style ring packings answer splendidly for machines running non-condensing, but are useless for running condensing, as the packing immediately burns up, causing no end of trouble to get the grooves clean again.

A large company in the north of England, which manufactures turbines, tried a water gland of the paddle-wheel type. It practically consisted of two grooved collars shrunk on to the shaft with a paddle wheel between them. This idea was not given a fair chance, however, the result being that the whole thing was put in the lathe, the wheel turned off to a level with the two collars, then it was grooved and turned into a steam-packed gland. When these glands were put into service the fun began. The first one went on the machine that changed from non-condensing to condensing. The temperature being kept up by the sealing steam, the collars did not follow the contraction of the spindle; consequently, they shifted on the shaft and started to rub. This added more heat, which helped to increase the expansion, resulting in showers of brass from the strips which were calked into the gland, at the same time setting up heavy vibration.

Another one went in on starting up. The machine had been well warmed up and the glands sealed to pull a vacuum, with the result that when the machine had been running a few minutes both glands fired. Anyone who has not had the misfortune to witness a gland properly fired has no idea what

a brilliant display it makes. One thing is certain: If there is any loose part in the machine it will soon make itself evident.

These cases simply show one of the drawbacks to steam-sealed glands, especially when using live steam with a gland simply shrunk on.—"Power."

MODERN APPLICATIONS OF SUPERHEATING TO MARINE STEAM BOILERS.

Paper read by ARTHUR SPYER, M. Inst. C. E., at the Engineering Conference of the Institution of Civil Engineers, June 21, 1907.

In dealing with this subject in a Conference note, it is only possible to make brief allusion to some of the interesting facts at disposal.

Information on the behavior of superheated steam as a gas is still deficient, and some recent experiments carried out by Messrs. Knoblauch and Jacob, at Munich, show that the specific heat at any given pressure is considerably higher than hitherto assumed; that it diminishes as the amount of superheat increases up to a certain point, and then rises again; and these results will influence theoretical calculations made in connection with the subject. [Similar experiments are being carried out at the National Physical Laboratory, and they confirm the German results.]

Superheaters which have been applied for marine work may be divided into two classes:

1. Those fitted in the uptake, and absorbing the heat from the gases of combustion after leaving the boiler.

2. Those placed in the boiler in such a position as to absorb heat from the gases during their passage through the boilers.

As to the first type, the largest applications for marine purposes in this country have been made by Messrs. Thomas Wilson, Sons & Co., who in 1900 fitted the *Claro* with superheaters. These superheaters consist of a series of multiple

U-tubes, with collectors at the inlet and the outlet. The result obtained in the *Claro* was so satisfactory that the system has been largely extended in the Wilson fleet, and they have at the present moment twelve vessels fitted or being fitted with superheaters.

Owing to the low gas temperature available a large superheater surface has to be provided, ranging from 40 per cent. to 50 per cent. of the boiler-heating surface, in order to give a superheat of 90 degrees to 140 degrees.

As regards results, the *Aleppo*, which has been fitted with superheaters, and runs with an average of about 150 degrees of superheat, has been carefully compared as regards coal consumption on her voyages with two sister ships, identical in other respects, running on similar service and at the same speed, but not fitted with superheaters; and it has been found that the coal consumption of the *Aleppo* is 12½ per cent. less than the sister ships over several years of service.

In the case of the *Martello*, which is fitted with Babcock & Wilcox boilers, the result has been somewhat obscured by the fact that when superheaters were fitted new cylinders were also fitted, and the cylinder proportions were considerably modified from those originally obtaining. The saving was remarkable: the vessel now uses only three of her four boilers, she runs at the same speed on the same service as she did previously, and the coal consumption has been reduced from 44 tons to 31 tons per day. The amount of superheat used in the *Martello* is 90 degrees, the surface of the three boilers is 8,055 square feet, and the superheater surface 2,000 square feet.

One of the most important questions is the condition and material of the working surfaces of the cylinders and slide valves. On this point Messrs. Wilsons' experience is of great value, as it is found that there is no trouble whatever. A very small amount of lubrication is applied internally and on the piston rods. The oil used has the specially high flash point of 700 degrees, does not form an emulsion in the feed water, and is easily removable by an ordinary mechanical filter.

As regards material of the rubbing surfaces, cast iron is invariably used; piston valves are fitted in the high and intermediate cylinders and flat valves in the low-pressure cylinders, and it is found that the wear is in many instances not so much as in ordinary cases. Moreover, there is much less trouble with the piston rods and glands than where saturated steam is used, and there is less leakage from pipe joints, etc.

As regards durability, a tube was cut from the *Claro* after five years' service, and no wear could be detected in it; and recently I had a tube cut out from a superheater which has been at work in a boiler on shore for eleven years, and failed to detect any wear in it. This tube is on the table.

The Central Marine Engine Works, of Hartlepool, have also installed several superheaters of type No. 1, one case being that of the *Inchmona*, in connection with quadruple-expansion five-crank engines, and a boiler pressure of 225 pounds per square inch.

The superheaters are fitted directly in the uptakes, below the air-heating tubes, and consist of a series of spiral coils, connected at top and bottom by cast-iron headers. The results obtained in the *Inchmona* were so encouraging that the system was repeated in the *Inchdune*, *Inchmarlo* and *Nasovia;* and it is stated that the *Inchdune* and *Inchmarlo* have been able to run whole voyages on an average consumption of 1 pound of coal per indicated horsepower hour.

A vessel called the *Teutonic* has also been fitted with superheaters, the superheater in this case consisting of a series of multiple U-tubes.

The economy directly attributable to superheating, from the data in the possession of the Central Marine Engine Company, works out to about 8 per cent.; but it should be noted that the total degree of superheat imparted to the steam is only about 70 degrees. In the experience of the Central Marine Engine Company, no ill effects have been noticed in the working of the engines due to the use of superheated steam.

As regards the superheaters of the second type—namely,

those placed in an intermediate position in the boiler—the only one which has been largely used is the Babcock-Wilcox. There are isolated cases of other systems, such as the Pielock superheater, which has been adopted in an installation of ordinary cylindrical marine boilers in France, in a vessel called the *Larence.* In this system the superheater is arranged in the water space of the boiler, and consists of a steam and watertight casing enclosing the heating tubes through a portion of their length ; the fire gases flow through the tubes of the boiler, thereby superheating the steam which passes round the portion of the tubes inside the casing.

The Babcock-Wilcox superheater is applied in conjunction with the Babcock-Wilcox marine boiler, in which the gases are passed three times through the boiler-heating surface by means of suitable baffling, and the superheater is placed between the first and second passes. By this means a gas temperature is obtained sufficient to impart a good degree of superheat with a moderate surface in the superheater.

The first tests with this superheater were made in America, on a vessel called the *J. G. Wallace*, fitted with boilers of 5,800 square feet and superheaters of 830 square feet of surface. The superheaters were placed in the first pass, but arranged so that by a simple alteration of baffling they might be cut out of the circuit altogether ; thus a comparison could be made using the same boilers in the same ship, and without superheaters. The result of these trials is given below :

	Saturated steam.	Superheated steam.
Indicated horsepower	1,530	1,559
Coal per indicated horsepower, pounds	1.76	1.51
Average steam pressure, pounds per square inch	238.7	237.6
Main feed temperature, degrees Fahrenheit	192.7	195
Revolutions per minute	77	79

From this it appears that the increased coal consumption due to using saturated steam was 16½ per cent.

A curious thing occurred in these two trials. It was found that when using saturated steam the power was developed with a somewhat earlier cut-off. The reason of this is not

very clear, because the same volume of superheated steam should contain sufficient heat to develop the same power, although, of course, the weight would be very considerably less.

Following on the *J. G. Wallace*, the American Navy made an experiment by fitting superheaters to four out of the eight boilers of the *Indiana*. The results were not published, but that it was favorable is certain from the fact that since then superheaters have been ordered for four other American warships.

There are now under construction two sister ships of the *J. G. Wallace*, fitted with superheaters, and also a vessel named the *Creole*. This latter, when complete, will be of special interest, as the vessel is fitted with Curtis turbines, working at 250 pounds pressure. The principal particulars of this installation are as follow:

Indicated horsepower	7,500
Heating surface in boilers, square feet	28,150
Heating surface in superheaters, square feet	4,350
Grate area, square feet	783
Number of boilers	10

Recently the Admiralty decided to fit superheaters to six of the boilers in H. M. S. *Britannia*. A boiler fitted with superheater was erected in the test house at Messrs. Babcock & Wilcox's works at Renfrew, and four trials were made at rates of coal consumption varying from 14 pounds to 30 pounds per square foot of grate per hour; and a noticeable point in the trials was the constancy of the superheat, which varied only from 75 degrees to 91 degrees. This is of importance, as for navy purposes the rates of working vary considerably; and it is therefore necessary that the superheaters should be so designed that the superheat obtainable should be as nearly as possible constant.

A series of tests were carried out by the Admiralty on the official sea trials of the vessel. The first two trials were of special interest, as in one case six boilers were used without superheaters, and in the other case six boilers with superheaters. The results obtained are as follow:

	Six saturated steam boilers.	Six superheated steam boilers.
Indicated horsepower	3,521	3,410
Coal per indicated horsepower, pounds	2.07	1.77
Water per indicated horsepower:		
Main engines, pounds	16.92	13.63
Auxiliary engines, pounds	4.1	4.56
Total, engines, pounds	21.02	18.19
Steam pressure in boilers, pounds, per square inch	196.3	200.0
Superheat at boilers, degrees	...	92½
Superheat at engines, degrees	...	83
Feed temperature, degrees	72.0	65.6
Water per square foot of heating surface actual, pounds	5.5	5.17
Coal per square foot of grate, pounds	17.6	14.5
Temperature in uptakes, degrees	398	347

In reference to these trials a curious feature appeared. Apparently the water consumption in the auxiliary engines was increased from 4.1 pounds per indicated horsepower using saturated steam, to 4.56 pounds per indicated horsepower using superheated steam. The reason for this is not clear, as the auxiliary engines, which are most of them of the simple expansion type, are just those which should show a considerable gain in economy when using superheated steam.

A trial was also made at 7,000 horsepower with the same two groups of boilers at work, one set on the starboard engines, the other on the port engines. The engine water consumptions were 13.26 pounds per indicated horsepower with superheated steam, against 15.01 pounds with saturated steam, each engine developing 3,500 horsepower. The uptake temperature recorded was 346 degrees for the superheater boiler and 370 degrees for the saturated-steam boilers.

On the subsequent trials of the *Britannia*, at 13,000 horsepower and at her full power of the 18,000 horsepower, the boilers with superheaters showed a lower uptake temperature than those without superheaters; at 13,000 horsepower the difference was 29 degrees, and at 18,000 horsepower it was 7 degrees.

Experience, therefore, would appear to show that with reciprocating engines, when properly adapted for the use of

superheated steam, a high degree of economy is obtainable, and that no detrimental results arise from its use. There is apparently no reason, seeing that turbines can be constructed to use superheated steam satisfactorily for land purposes, why they should not also be constructed to use superheated steam for marine purposes; and developments in that direction would appear to be the next step, as already indicated in the American vessel *Creole*, with Curtis turbines.

There are, I believe, some other owners who have used superheaters for marine work, but there is, unfortunately, not any information available as to the experience obtained with them; and I can only hope that this brief indication of some of the results actually obtained with superheated steam for marine purposes may be of interest.

A NEW TORSION METER.

By B. HOPKINSON and L. G. P. THRING.

With the increasing use of turbines for marine propulsion the problem of directly measuring the power delivered to a shaft by the engine has become of pressing importance. Several instruments for this purpose have been designed and have proved more or less satisfactory, the best known being that of Messrs. Denny & Johnson, which is now considerably used. All these instruments measure the twist in a length of the shaft, whence, knowing the rigidity of the shaft and the speed of revolution, the horsepower transmitted can be at once deduced. In Messrs. Denny & Johnson's torsion meter, and in the others with which the authors are acquainted, it is necessary, in order to secure reasonably accurate results, that a considerable length of shafting be taken. In Messrs. Denny & Johnson's apparatus two wheels are clamped to the shaft as far apart as is conveniently possible, and the relative displacement of these two wheels produced by twisting of the shaft is measured by an electrical device. According to figures given by Mr. Archibald Denny at the meeting of the

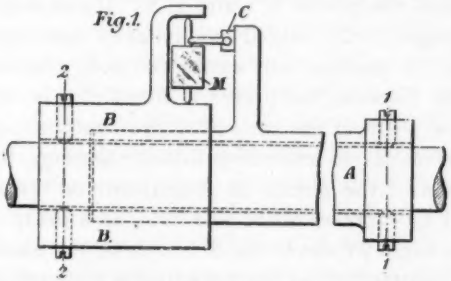
Institution of Naval Architects in March, 1907, the displacement at the periphery of the wheels can be determined correct to $\frac{1}{100}$ inch, a remarkably good result, considering the nature of the means employed. If we assume that the wheels are three times the diameter of the shaft, and that the shear in the latter at its periphery, when running fully loaded, is $\frac{1}{2000}$, the corresponding displacement at the surface of the wheels, if placed 50 feet apart, will be $3 \times \frac{50}{2000} = .075$ foot, or $\frac{9}{10}$ inch. Thus, under these circumstances, an error of $\frac{1}{100}$ inch in measuring the relative displacement of the two wheels amounts to about 1 per cent. on the full-load twist of the shaft. If a length of only 10 feet were available, the corresponding possible error would be 5 per cent. It may be assumed, therefore, that a length of the order of 10 feet is the minimum which will give good results with this form of apparatus; and Mr. Denny, at the meeting referred to, stated that they rarely made use of less than 15 feet or 20 feet of shafting. This requirement, while not militating greatly against the use of the apparatus in big ships where there is plenty of room round the shafting, and considerable lengths are easily accessible, seriously restricts its application in small vessels.

It should be observed that with a given ratio of wheel diameter to shaft diameter, the length of shafting required to secure a given percentage of accuracy is independent of the diameter of the shaft; for all shafts when fully loaded are sheared by approximately the same amount at the periphery. Thus two points a given distance apart on the surface of any shaft will be relatively displaced by a constant amount, independent of the diameter of the shaft, which may be taken as about one two-thousandth part of their distance apart. It is this relative displacement, magnified in the ratio of the wheel diameter to the shaft diameter, which is really measured by these torsion meters.

In the authors' apparatus, described in this paper, the length of shaft in which the twist is measured may be reduced to about 12 inches, the twist being measured correct to 1 per

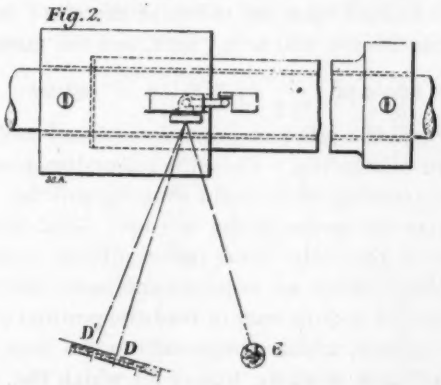
cent. of the maximum which the shaft can transmit in normal working. Figs. 1 and 2 show one of the earliest forms that were tried; it was subsequently altered considerably in detail, and is given here only because it is easy to follow the principle in it. The sleeve A is clamped to the shaft by screws

Fig. 1.



in the plane 1, 1. The collar B is similarly clamped in the plane 2, 2. On the collar is mounted a mirror M which is pivoted in a frame carried on the collar, so that it can turn about an axis at right angles to that of the shaft. The axis

Fig. 2.



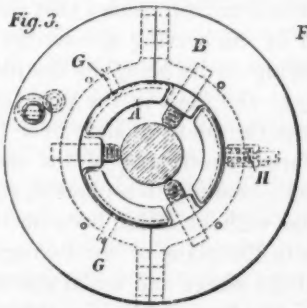
of the mirror carries a short arm which engages with a flat plate C carried on the free end of the sleeve. Twist of the shaft between the planes 1, 1 and 2, 2 causes the plate C to move relative to the supports of the mirror, thus tilting the latter about its axis. When the shaft is in the position shown,

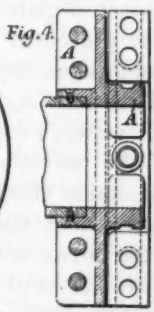
the mirror, which is concave, forms (on a ground-glass screen) at D an image of a straight-filament lamp placed at E. When the shaft twists so that the mirror turns, this image is displaced to D′ through a distance which is proportional to the relative motion of sleeve and collar, or to the twist in the shaft between the planes 1, 1 and 2, 2. If the shaft is revolving, the image is still formed momentarily once in each revolution, and its position can easily be seen. Assuming that the distance between the planes 1, 1 and 2, 2 is 12.5 inches, and that the shear at the surface of the shaft is $\frac{1}{2000}$ (which about corresponds to normal full-load running), the relative displacement of the points of attachment of the sleeve and collar will be 0.00625 inch, and this, as already explained, will be the same whatever the diameter of the shaft. In the apparatus constructed by the authors the distance of the actuating plate C on the collar from the center of the shaft is 2.9 times the shaft radius. The movement of the plate will then be 2.9 times that of the surface of the shaft, or 1.8 hundredths of an inch. The corresponding angular displacement of the mirror will depend upon the radius of the arm; in the apparatus, as constructed, this is 0.3 inch, and the mirror is tilted through an angle of $\frac{0.018}{0.3} = 0.06$ of a radian. The angle through which the reflected beam is displaced will be double this, or 0.12 of a radian. Thus the linear displacement of the image corresponding to full-load running will be 0.12 of the distance from the screen to the mirror. Good definition can be obtained if the latter is 60 inches, giving a displacement of 7.2 inches. With an ordinary spectacle lens, silvered to form a mirror, it is quite easy to read the position of the image correct to $\frac{1}{20}$ inch, which corresponds to less than 1 per cent. of the maximum working torque for which the shaft is designed.

It is evident, therefore, that so far as magnification is concerned, the twist in a short length of shaft can be read quite sufficiently accurately by this method. It was found, however, that in the form of apparatus described above, when

used on small shafts, the bearing between sleeve and collar was apt to be strained in putting the instrument on or by bending of the shaft. Friction was thus set up, which caused error in reading, partly from the resulting deformation in the instrument, and partly because the shaft was relieved of part of the torque which was transferred to the instrument. The attempt at a complete constraint between the two members was therefore abandoned, and a partial constraint was substituted which permitted motion in certain ways, and so practically eliminated friction. At the same time the mirror and actuating plate were so disposed that relative motion of sleeve and collar other than pure twist produced no motion of the mirror. Such motions necessarily occur in ordinary working, in consequence of bending of the shaft, and, as the motion of the actuating plate is only about $\frac{1}{50}$-inch, they may cause serious errors unless eliminated. The manner in which this result has been secured will be apparent from the detailed description of the apparatus as used, which follows.

In the engravings, Figs. 3 to 5, A is the free end of the sleeve, and B the collar, each being made with deep flanges for stiffness. They are split into halves, so that the appa-

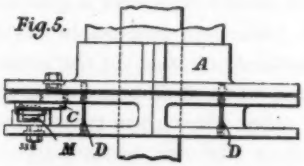




ratus can be fitted over the shaft when in place. Three screws with rounded ends are used for clamping: one set at the end of the sleeve, and another set passing through the central plane of the collar. An extension of the sleeve passes through the collar and engages with the round ends of the screws G,

G carried on the collar, being held up into engagement with them by means of a spring H. The end of the sleeve is otherwise quite free. The elasticity of the sleeve and the slight "give" in its clamping screws are sufficient to allow of its coming to a bearing on the screws G, G, without much pressure on the spring H, even when the shaft is slightly bent. The mirror is carried in a frame fixed to the flange of the

Fig.5.



collar in such a manner that the mirror axis passes through the shaft axis. The mirror axis carries a small steel ball at its outer end, which engages with the plate C. This latter is fixed to an arm carried on the sleeve flange and passing through a hole in the collar flange, and its face is parallel to the shaft axis. The ball on the end of the mirror is approximately (in theory it should be exactly) in the plane of the screws G. With this arrangement it is obvious that any relative movement of plate and mirror, except that due to twist, will either be prevented by the bearing against the screws G, or will be innocuous, owing to its being in the plane of the plate C. Thus bending of the shaft in the plane of the mirror axis will merely cause the plate to slide over the ball at the end of the mirror arm; shortening of the shaft due to thrust has a similar effect; bending in a plane at right angles to the mirror axis is also without effect, because the ball at the end of the mirror is in the plane of the two screws G, G. When the two halves of the sleeve and collar are taken apart they are kept in their relative position by the rods D, D, which, while stiff enough for this purpose, are sufficiently elastic not to affect the reading when the apparatus is on the shaft.

The apparatus was first tested statically on a shaft of the same diameter, and probably of the same material, as the tur-

bine shaft upon which it was ultimately to be used. This shaft was $3\frac{3}{8}$ inches in diameter, and was mounted in the works of Messrs. J. I. Thornycroft & Co., Limited, with arrangements for applying a torsional load equal to about half that which it was to transmit when at work. It was found that the dynamometer reading was proportional to the load within 1 per cent. of the maximum applied throughout the whole range. The absolute twist in the shaft was also measured direct by pointers carried at the two ends. The modulus of rigidity for this shaft was thus found to be 11,800,000 pounds per square inch. There were slight signs of hysteresis, but these were not of any practical importance. The instrument was then fixed on the port wing shaft of the torpedo boat *Greenfly*, which is driven by the low-pressure turbine. The readings were taken during the Admiralty trials of the boat on March 25, 1907. On that day the boat did circling trials, and stopping and starting trials, so that the changes in power on the shaft were frequent and severe. With full steam in the turbine the shaft ran at 1,230 revolutions per minute, and the reading on the ground-glass screen, which was placed 57 inches from the mirror, was then 8.75 inches, corresponding to 1,100 horsepower. The deflection could be read easily correct to $\frac{1}{20}$ inch, or less than 1 per cent. The zero was determined by allowing the propeller to drag the turbine round with a good vacuum in the casing.

Two readings were taken, first with the boat going ahead, and, secondly, with the boat going astern, with moderate speed in each case. The mean of these readings, which differed by $\frac{4}{10}$ inch, were taken to be the zero. In the first half of the day's run the zero was found to have changed by 0.35 inch, corresponding to an error of 4 per cent. in the torque at maximum power. It is probable that this change was due to a slight shift of the scale, which was rigged up on packing cases in a very temporary fashion. Just before running home, after the trials, the zero was again carefully determined, and the boat then ran at full power for half an hour, care being taken to fix the positions of the lamp and scale. It was found

that during this run the zero had not shifted by so much as $\frac{1}{20}$ inch. Altogether the apparatus was running round for about eight hours, and was not touched in any way during that time, except once when the clamping screws were slightly tightened. It was brought down on the morning of the trial and fitted on to the shaft in about half an hour. The maximum length of shaft exposed in this boat is about 15 inches, and the clearance between the shaft and the vessel's plates is there about 8 inches; this is just sufficient to get the apparatus in. There is, of course, no independent means of checking the accuracy of the results obtained on a trial of this kind. But the facts that, when tested statically, the deflections were proportional to the twist of the shaft, and that when running the zero did not change, leave no doubt that the power readings when running must also have been correct to within 2 per cent., and probably within 1 per cent. of the maximum power transmitted.

The apparatus can be fitted on to a shaft of any size smaller than the bore, within reasonable limits; that designed for a $3\frac{1}{2}$-inch shaft works quite satisfactorily on a shaft of 2 inches diameter, and would, no doubt, work down to $1\frac{1}{2}$ inches. The over-all diameter is about $2\frac{1}{2}$ times the diameter of the largest shaft it will take. As regards length, the greater the length, of course, the greater the accuracy; but a length of 1 foot or 2 feet is ample. This length will suffice, whatever the diameter of the shaft, but the larger the shaft the easier is the accurate measurement of the power; because, while the linear displacements to be measured are the same, the disturbances due to bending and vibration become less.

The instrument, of course, measures the instantaneous value of the torque in the shaft at the moment when the reflected beam from the mirror strikes the screen. If the torque varies in the course of a revolution, as it will do with a reciprocating engine, it is necessary to measure it at several points and to take the mean. This can be done by using several mirrors placed at intervals round the apparatus, or by the use of more than one screen. In a turbine-driven shaft the torque will be

fairly uniform, though it will doubtless vary slightly with the position of the propeller blades in relation to the hull and the water surface.

Since carrying out the above-described tests we have learned that Mr. Hermann Frahm, of the firm of Messrs. Blohm & Voss, Hamburg, has been working at the same problem as that with which we have dealt, and has arrived, quite independently, though a little later in point of date, at a very similar solution. We understand that apparatus of the kind designed by Mr. Frahm has lately been tried on a German steamer, and has proved satisfactory. Mr. Frahm's apparatus is apparently the same as ours in principle, but differs from it in detail to a certain extent.

The apparatus as we have designed it can be easily applied to factory shafting, and will, we hope, prove useful for this purpose as well as in marine work, especially to those who have to put in electric motors for driving such shafting, and so require to know the exact power taken by a particular machine or set of machines.—"Engineering."

RECENT DEVELOPMENTS OF THE GASOLINE LIFEBOAT.

By George E. Walsh.

The evolution of the lifeboat has been coextensive with the growth of the life saving service of the leading nations of the world, and as no single country can claim a monopoly of patroling the coasts in the interests of humanity and property it is equally evident that the improvements in lifeboats is the result of the combined efforts of many nations instead of one or two. While the developments of new effective devices for naval fleets are held as close secrets by the different nations, and great jealousy is exhibited in protecting these improvements, the very opposite is noted in the work of increasing the efficiency of the world's life-saving service. Co-operation between the various nations is carried on extensively, and the discovery of a new plan for increasing the value of lifeboats or station equipments is quickly adopted by all.

England, America and France have led the world in the establishment of superior life-saving stations, lighthouses and other coast equipments for the protection of life and property on the high seas. Owing to their extensive and important stretches of sea coasts, it is natural that they should have taken the lead in this respect, and we look to them for the highest development of any particular feature of the work.

The lifeboat has received a greater amount of attention from these three nations than almost any other part of the service. Experiments have been conducted on a large scale by representative experts in the three countries, and while the size, shape and general construction of the boats may differ somewhat to meet local conditions, they are, in the main essentials, the same in the three countries. The need has always been of lifeboats that were not only unsinkable, but self-righting and self-bailing. A boat which could be tossed about on the waves like a cockle-shell and still preserve intact its usefulness is the only type of craft that is of any value to a life-saving station. The early type of lifeboat could often ride successfully the huge breakers on coasts, but if once capsized it was rendered useless, and likewise if filled with water by the waves it would become unmanageable, if it did not actually founder.

For some time now unsinkable lifeboats have been in use, and also boats which would automatically right themselves if capsized, and at the same time bail out the water thrown into them by the waves almost as fast as it could enter. With the lifeboat developed to such a point of efficiency there seemed but one other improvement possible. Dependent upon sail and oars for its propulsion, there were times when tide and wind would balk it from reaching a disabled ship, and repeated launchings in the surf were of no avail. If some power greater than that exerted by seamen at the oars could be installed in the lifeboats, the greatest difficulty would be overcome.

Within the last few years the problem has been solved simultaneously in England, America and France. The gasoline-

driven lifeboat is no longer an experiment, but a demonstrated actuality. Improvements in the power and range of effectiveness of the motors are still possible, but the practical use of the boats in the life saving service of the three countries indicates the future of this new type of craft.

It was difficult to install a lifeboat with a gasoline motor of the right size and capacity, and the problem involved a number of important considerations that grew more perplexing as the work proceeded. To attain anything like an ideal type it was necessary to retain present qualifications of the lifeboat, such as its ability to right itself after capsizing, bail out the water thrown into it, and be absolutely unsinkable, but at the same time possess other more intricate merits. The gasoline motor had to be placed in absolutely watertight compartments, and prove automatic in control and operation. In the event of being capsized the motor should automatically stop, and when righted again be ready to work as before.

One of the early successful adaptations of the gasoline motor to lifeboats in England was made at the Folkstone station in 1904. The experiments were tried that year with an old lifeboat, 38 feet in length and 8 feet beam. This boat was of the self-righting type, rigged with jib, fore-lug and mizzen sails, and double banks of twelve oars. She was equipped with air compartments to keep her from capsizing and sinking, but in placing a ten-horsepower motor in her some of the air cases under the deck amidships were removed. In place of the air cases a strong watertight compartment was built for the motor, pumps, vaporizers and electrical apparatus. The gasoline was carried in a metal tank forward and delivered by gravity to the engine.

This experimental craft proved so successful, especially in working to the windward of disabled ships and approaching them in a heavy sea, that a series of tests were carried on under the direction and supervision of the Royal National Lifeboat Institution. Improvements were finally made which developed a type of gasoline lifeboat which could make a speed of six knots an hour with a full crew and stores, and

when capsized by a crane the motor would automatically stop running and when released the boat would immediately right itself. When righted by its own bouyancy the motor started again by turning the starting handle. The improved lifeboat was placed in regular service at New Haven, and careful study of its actions in heavy seas has since proved its value.

Another type of lifeboat driven by a gasoline motor was exhibited at the Life-Saving and Hygiene Exposition at Paris in 1904 by one who was connected with the Government Arsenal at Rochefort. This craft represented a new feature by virtue of its peculiar movable keel. In order to secure great stability in any kind of seas the lifeboats of modern times have been equipped with heavy keels fixed to the bottom. In most of the English lifeboats an iron keel weighing upward of 500 to 650 pounds was usually attached to the bottom about twenty inches below the water line. Years of practical experience had demonstrated that the weight of this keel could not be increased much more without making the craft difficult of manipulation, and it was equally impossible to extend it downward much further without fatally handicapping the boat's operation on shallow coasts.

In the Henry type of lifeboat, with its movable keel, the attempt was made to increase the boat's stability without decreasing its efficiency in either of the above particulars. The movable keel consisted of a torpedo-shaped cast-iron weight of 650 pounds. This keel was movable so that when the boat entered deep water it could be extended downward, and when approaching land the keel could be raised, or if it struck an object it would ascend of its own accord. The movable keel enters a chamber in the bottom of the boat, and when in this position the draught of the boat is only two feet, but when fully lowered the draught is increased to five feet. The result of this device is that the boat has remarkable stability in a rough sea and is most difficult to capsize. In practical tests the lifeboat, with the keel lowered, safely breasted waves which would swamp and capsize almost any other type of craft.

This same boat was equipped with sails, oars and a small motor. While the motor was primarily intended as an auxiliary power it has proved its efficiency in storms and rough weather, especially when approaching ships needing assistance. The danger of running up alongside of a ship in a heavy sea with a lifeboat propelled by oars and sail is easily imagined. With a small motor the management is made almost as simple in rough weather as in a quiet sea. The boat carrying 40 or 50 persons can make five or six miles an hour, and with only her crew and stores aboard she has developed eight miles by using the sail along with the motor.

In France the gasoline-driven lifeboat has reached a degree of efficiency that makes it of special value, and it has become an important factor in the life-saving service of that country. The first Henry lifeboat is now stationed at Brest, and others have been constructed on similar plans both in France and England. At the port of La Rochelle the experimental operations of this type of lifeboat were made a short time ago before delegates of different life-saving societies of the world.

In the United States the gasoline lifeboat has received in the last two years even more practical attention than in France and England. By virtue of our extensive sea coast, and the importance of shipping throughout its whole length, the lifeboat that can be efficiently managed in any kind of weather is of prime importance. The development of the type which has been adopted by the Government has been along lines similar to those of the English and French type, but with such modifications as local condition demand. None of the salient features of the improved lifeboat operated by oars and sails have been disregarded or weakened by the adoption of gasoline motors. The power boats are of the self-righting type, self-bailing and non-sinkable.

The American type of gasoline lifeboat embodies nearly if not all the best features of similar craft used in foreign services. The motor has been installed not alone as an auxiliary power, but as the main dependence for propulsion through rough seas to reach disabled ships. Oars and sails are also

provided, and the speed of the boat can be increased thereby, but the motor is powerful enough to secure great headway without any help from the crew.

The boat selected for the first test with gasoline-motor power has a hull 34 feet in length, 8 feet beam and 3 feet draught. It is rendered unsinkable by three cross airtight bulkheads and two longitudinal ones below the water line. These bulkheads are filled with 82 copper air tanks, so that the danger of sinking from a collision is reduced to a minimum. Half of these air tanks are sufficient to make the craft unsubmergible. To prevent the boat from capsizing there are two additional air chambers, one at the bow and the other at the stern, and these two extra chambers are sufficient to support the craft without the 82 copper tanks below the water line. The combined buoyancy of these numerous air chambers is enormous, representing upward of 12 tons. In actual tests it required 44 men to force the boat over on one side so that the deck scuppers were awash.

But if water should enter the boat, either through excessive leaning or by the breaking of huge waves over it, there is no danger of swamping. The craft is provided with abundant devices for self-bailing. A series of ten six-inch copper tubes, five on either side, run along the inside of the deck. Water shipped over the rails or when the boat is on its beam ends escapes through these tubes within a few seconds, and, as they are provided with automatic valves to shut off incoming water from the other direction, the craft can bail itself out continually as fast as the waves pour into it.

The American lifeboat is not equipped with the movable keel of the Henry type, but the placing of the heaviest weight below the center of gravity accomplishes the same end. The gunmetal keel of 1,050 pounds, the metal centerboard of 750 pounds and the copper air cases of 900 pounds weight are so placed and adjusted that the total weight of 2,700 pounds acts as a great equilibrium factor and makes the boat remarkably stable. Lateral stiffness is obtained by the large movable

centerboard, the keel and the long, flat floor. Great stability in small craft is obtained in this way without a movable keel.

The gasoline motor is of 20 horsepower, of the four-cylinder, auto-marine type, and drives an 18-inch propeller at 400 revolutions per minute. The motor is placed in the after air chamber, and is protected by a watertight door in the bulkhead. It is easy of access, but completely shut off from all air and water when in operation. Sufficient air is supplied for the purpose of completing combustion of gases by an ingenious pipe which runs direct to it from the outside. There are two fuel tanks in the forward chamber. The larger, of 75 gallons capacity, is placed in the lower part of the boat, and the smaller, of 25 gallons, above it. The fuel is pumped from the lower to the higher, and then fed to the motor from this by gravity.

The automatic control of the motor is simple and effective. When capsized the motor automatically stops, and when righted again it is immediately started by turning a crank. This control device consists of two pairs of rings mounted in a vertical position. The lower halves of the rings are of metal and the upper halves of some insulating material. A metal ball rolls freely in these rings, and makes the necessary electrical connection to cause ignition of the gas. So long as the boat is in an upright position the balls maintain constant electrical connection, but the moment the boat is upset the balls roll into the upper halves of the rings, which are composed of insulating material; electrical connections are immediately stopped and the motor ceases to operate. When the craft is righted the ball falls back into the metal halves of the rings, and it requires merely a turn of the starting crank to make the motor work.

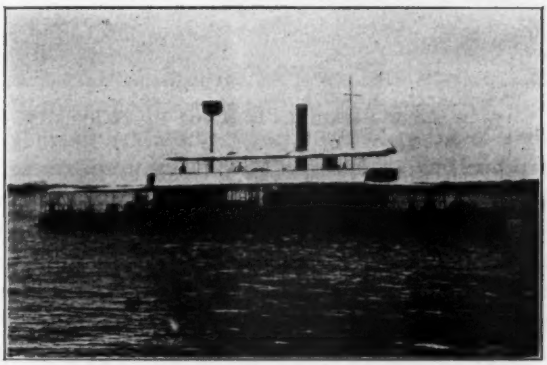
The dangers of a motor not automatically controlled when the boat is capsized in a rough sea are many. If not stopped it might get tangled in the rigging and kill some of the crew, or it would drive the boat away from the men so they could not regain it. Such control is absolutely essential to the successful operation of the modern lifeboat.

The lifeboat is steered by a rudder worked through a spur-wheel and curved rack, but it can instantly be disconnected and the rudder hoisted out of the water so that the craft can be steered by oars. Often in approaching the shore in breakers a rudder is dangerous, and steering by an oar is essential. Provision is thus made for almost any emergency.—"The American Marine Engineer."

SHIPS.

BRAZIL.

New Gunboats for the Brazilian Government.—The gunboats which we herewith illustrate are two of the seven gunboats—four of the larger and three of the smaller—recently constructed by Yarrow & Company for the Brazilian government for service on the Amazon. The dimensions of the four twin-screw shallow-draft gunboats are: Length over all, 120 feet; beam, 20 feet; depth, 4 feet 9 inches. With a load of 25 tons they draw 25 inches, and have a speed of 12½ statute miles an hour. They are fitted with Yarrow's patented hinged flap, by means of which it is claimed that one mile an hour additional speed is obtained, without additional power or additional cost.

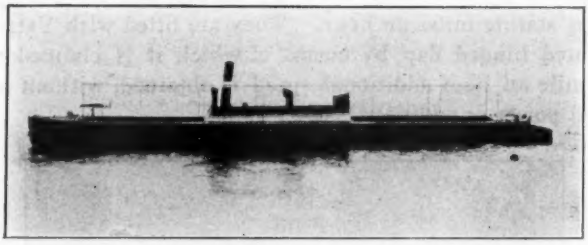


TWIN-SCREW SHALLOW-DRAUGHT BRAZILIAN GUNBOAT.

Two deck houses are provided, these being constructed of steel, one forward and one aft. The tops of these are connected by the battery deck with steel bulwarks all round,

and over this awnings extend from forward to aft, as will be seen in the engraving. The forward cabin is divided into two compartments, that forward being fitted up for the commander and the other arranged as a wardroom. The after cabin is fitted up as a petty officers' quarters, and above is provided a hollow steel mast, with crow's nest at the top. Above the forward cabin is the conning tower and the steering position, and above these again a searchlight platform.

From a structural point of view, the feature of the design is the method by which longitudinal strength is obtained for these shallow lightly-constructed vessels. It lies in the connection by the upper steel deck uniting the two deck houses, the bulwarks of this upper deck forming a girder throughout the amidship portion of the vessel.



SINGLE-SCREW PATROL LAUNCH FOR RIVER SERVICE.

The propelling engines are triple-expansion, these being supplied with steam by a Yarrow water-tube boiler. The auxiliary machinery includes steam steering engine, distilling apparatus, circulating engine, electric-light plant, fans and pumps. A steam capstan and warping gear is fitted forward, with the necessary anchor arrangements, and the usual ship's boats are provided. These vessels were shipped in pieces, and were riveted up and launched on the banks of the Amazon.

The dimensions of the three smaller single-screw gunboats are: Length, 75 feet; beam, 9 feet 3 inches; depth, 4 feet. They were also constructed on Yarrow's tunnel system, with patented adjustable flap.

As will be seen in the illustration, the machinery space is completely inclosed. The machinery and steering station are protected by chrome steel plates, proof against the Lee-Metford rifle fire point blank at 20 yards. The boats are fitted with two rudders, and can be steered either from forward or from a sheltered position aft of the machinery space. A wooden awning is provided, extending from end to end, and there is a short deck forward, on which, it will be seen, is placed a small gun.

This type of launch is now attracting considerable attention, because of its suitability for expeditionary work, its characteristic features being the extreme simplicity of the machinery, the engine being non-condensing, and the boiler of the locomotive type, which is proportioned for burning wood. The draught, with steam up, is 14 inches, and on this draught a speed of 10 statute miles an hour is easily maintained. These launches were shipped entire from London, and are considered to be just what is required to keep natives under control on some of the rivers which pass through more or less unexplored districts.

ENGLAND.

In the matter of Injuries to Ships, fate seems to be working as hard as the Government in the direction of reduction. The following is a not necessarily complete list of ships that have been ashore or badly damaged in collision during the last year or so: *Donegal*, *Good Hope*, *Dominion*, *Commonwealth*, *Montagu*, *Mars*, *Duncan*, *Albemarle*, *Prince George*.

Of these the *Good Hope* ripped up a good deal of her bottom; but as she was doing nearly 24 knots recently, her injuries must have been of a very temporary nature. The *Donegal* got off less lightly, and will probably always be something of a lame duck. The *Dominion* certainly will be, and the *Commonwealth* is likely to be non-effective for many months. The *Montagu*, of course, has gone for good. The *Mars* scraped her bottom at the Needles entrance to the Solent; no information is forthcoming as to her present con-

dition. The *Duncan* grounded at the time of the *Montagu* salving, and probably did herself no good. The *Albemarle* has been reported very seriously injured by the recent collision; but from all accounts, her name was used in error for the ship that she hit. Her real injuries are believed to be small. The *Prince George* is, or ought to be, used to accidents by now, and seems to have been unhurt by her recent hitting of a sand bank.

H. M. S. Bellerophon.—The latest first-class battleship which has been built for the British Navy, the *Bellerophon*, was launched from the Royal Naval Dockyard at Portsmouth, July 27. The *Bellerophon* is a sister ship to the *Dreadnought*, but is slightly larger. Her displacement is 18,550 tons, as compared with 17,900 of the *Dreadnought*, and she draws half a foot more water. In other respects her chief dimensions are the same as those of her sister ship, that is, her length is 490 feet, beam 82 feet and draught 27 feet. She is to be fitted with turbine engines of 23,000 horsepower, which are being built at Fairfield. It is estimated that her speed will be 21 knots. We believe that the *Bellerophon* will be similarly armed to the *Dreadnought*, with the exception that she will carry 4-inch guns to resist torpedo attack, instead of 12-pounders. The rapid progress made in the construction of the ship may be gathered from the fact that she was only laid down on December 3d, 1905, and it is hoped that she will be ready for commissioning by the end of this year.

H. M. S. Dreadnought.—The accompanying tabular statement gives the results of the last progressive speed trials of the *Dreadnought*. It was intended to make forty runs, but the new propellers were so obviously unsuitable to the conditions that the experiment was abandoned after twenty-eight runs had been completed. Using propellers on one side only, five degrees of helm kept the vessel steady on her course. It will be observed that the port engines, running alone, at 297 revolutions per minute, giving 12,500 shaft horsepower, had a shaft torque approaching that given by both sets of engines

running at 341 revolutions, giving combined 27,500 horse-power.

Both the old and new propellers are of the "cavitating" species, taking large power and giving small thrust, and the direction in which improvement may be hoped is not difficult to see. As the positions of the propellers fore and aft and athwartship cannot be altered, there only remain alterations in dimensions and forms of bosses, diameter, pitch, surface and number of blades and their thicknesses. And it is well to try propellers at deep-load draught. It is well to keep in mind that the ship was about 1,150 tons heavier than she was on her acceptance trials.

H. M. S. DREADNOUGHT.

Revs. per minute.	Speed, knots.	Torque.		Shaft horsepower.	Approximate.	
		Inner shafts.	Wing shafts.		Revs. per knot.	Slip per cent.
		Original Trials.				
132	9.0	...	...	1,750	880	17.5
165	11.0	...	...	3,500	900	19.3
298	19.4	.6	.52	17,700	922	21.2
337	21.6	.9	.6	26,900	936	22.4
341	21.8	.98	.6	27,500	938	22.5
		Recent Trials.				
215	14.6	.45	.2	7,000	884	17.8
294	19.0	.8	.4	18,000	928	21.6
330	20.7	.93	.6	26,400	957	24.0
S { 233	12.0	.66	.38	6,435	1,165	37.7
S { 290	15.5	.91	.56	11,500	1,122	35.4
P { 248	13.0	.68	.38	7,100	1,145	36.5
P { 297	15.1	.94	.64	12,500	1,180	38.4

S = Starboard engines only.
P = Port engines only.

The Dreadnought.—*Reduced Speed at Full Draught.*—A correspondent, whose position enables him to speak with authority, writes: When the *Dreadnought* carried out her full-speed trials she did 21 knots on a draught of 26 feet. On her arrival at Villagarcia, after she had completed with stores, her draught was 30 feet. It had been calculated that the

reduction in speed on completing with stores would be a ¼ knot; instead of that the speed lost was 1½ knots. On her full draught the ship is not a 21, but a 19½-knot ship. At present the officers cannot control her at 15 knots, and, therefore, under this speed she is impossible as a station keeper.

New Type of Armored Cruiser.—It is understood that the Admiralty have decided to make provision in the next shipbuilding programme for the laying down of a new type of armored cruiser at Pembroke Dockyard during 1908–9. In size she will range between a second-class cruiser and the ships of the *Duke of Edinburgh* class. The vessel will be protected by side armor, and will mount 7.5-inch guns. She will have a speed of 23 knots per hour, and her relatively large bunker accommodation will render her, with the features above mentioned, a suitable vessel for detached police duties in remote stations.

Boadicea.—The unarmored cruiser *Boadicea*, recently laid down at Pembroke Dockyard, will have a length of 385 feet; breadth, 41 feet; load draught, 13 feet 6 inches, and displacement 3,300 tons. She will be fitted with turbine engines of 18,000 H.P. Her armament will consist of five 4-inch breech-loading guns. She will have double bottoms constructed to carry oil fuel, and a large bunker capacity.

Tartar.—The torpedo-boat destroyer *Tartar*, ordered from Messrs. John I. Thornycroft & Co., Ltd., has been successfully launched from the company's Southampton works. The dimensions of the vessel are: Length, 270 feet; beam, 26 feet; draught, 8 feet 2 inches. The contract speed is 33 knots. The vessel is to be fitted with turbine machinery of the Parsons type (built by Messrs. Thornycroft) and six Thornycroft water-tube boilers, arranged for burning oil fuel. The armament will consist of three 12-pounder quick-firing guns and two 18-inch torpedo tubes.

H. M. S. Cyclops.—This vessel has now been fitted to act as a floating dockyard. On her lowest deck is a fully-equipped foundry and forge, with cupolas, where damaged parts of machinery, can be replaced by new castings. Then there are

carpenters', blacksmiths' and armorers' shops fully equipped, fitting works, coppersmiths' and electricians' departments. On a higher deck is a boiler shop, where boiler and ship plates can be dealt with, punching and shearing machines being there just as in a land shipyard. A powerful crane travels all round the ship to lift repairs from the holds and on or off the warship that has come up for repairs. An electricity-generating station is also included in the *Cyclops*' equipment for by this power all the machines and cranes are worked. Ice-breaking plant is also carried, and, as a contrast, refrigerating machinery is also represented, whilst on another deck is a gigantic set of condensers, capable of supplying a whole fleet with fresh water. A crew of 300 men, mostly artificers, will be required on the vessel, which is believed to be the only specially-built craft of its kind in the world.

FRANCE.

French Submarines.—All the submarines actually in service in the French Navy are propelled entirely by electricity, and have but a limited radius of action. This is their principal fault. The radius of action of the earlier types only allows of a journey of from 30 to 40 miles from the electrical charging station. In addition to this the accumulators are subject to rapid deterioration; they require careful management, take up a lot of space, and are very heavy. All these added together limit the employment of this type of boat.

The speed is, as a rule, from 8 to 10 knots, and in order to remedy the slow speed and a limited radius of action, the types most recently launched or being constructed have been considerably modified. The submarine has been made more closely to approximate to the submersible, and in the near future they will form only one class. The submarines of 320 tons displacement will be provided with petrol motors of 600 H.P., capable of giving them a speed of 12 knots.

The French Submarine Gymnote recently sank as she was leaving the docks at Toulon. One is not altogether surprised at the fate which has overtaken this boat. From the very first

her behavior has been erratic, and she has met with more than one mishap previous to her last exploit. The incident is a reminder how much has still to be learnt with regard to the design and construction of submarine boats in order to insure safety and stability under service conditions. She is a single-screw steel submarine boat, designed by *Gustave Zédé*. Her length is 56 feet 5 inches; diameter, 5 feet 11 inches; displacement, 30 tons; I.H.P., 55=8 knots surface speed, 6 knots submerged speed. She carries detachable lead keel. The motive power is electricity, derived from Laurent-Cély battery. Radius, 32 miles at 8 knots, 100 miles at 4 knots. Her armament consists of two 335-mm. torpedoes, carried in drop collars, one on each beam.

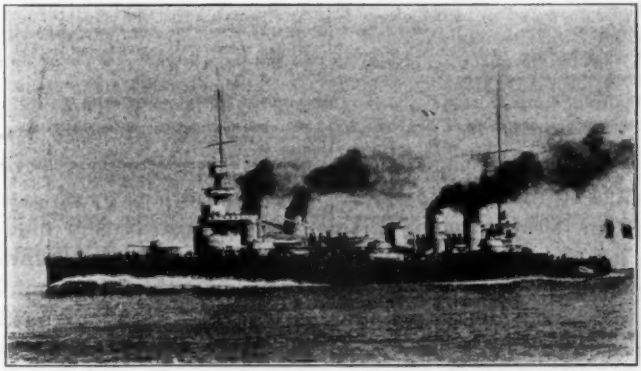
Gambetta Class.—All three cruisers of this class have now done their "reliability trials." The results are as follows: *Victor Hugo*, 96 hours—average, 19.5 knots; *Jules Ferry*, 72 hours—average, 20 knots; *Leon Gambetta*, 24 hours—average, 19 knots. At the end of twenty-four hours the *Gambetta* did a couple of hours at full speed, and made 22 knots.

The French Battleship Patrie recently burned her funnels out. According to the semi-official "Moniteur de la Flotte," Messrs. Niclausse will have to pay for this. Were a similar rule enforced in this country some boilermakers would find contracts far from profitable, as funnels burned out more or less are by no means uncommon now-a-days after hard steaming bouts.

French Cruiser Victor Hugo.—The third of the larger French armored cruisers, the *Victor Hugo*, has recently undergone her official trials. Her sister vessels are the *Leon Gambetta* and the *Jules Ferry*.

The figures which we are about to give are interesting as affording a comparison between two vessels of similar dimensions and provided with engines of the same type and calculated power, but having different types of boilers. Thus the *Leon Gambetta* had Niclausse and the *Victor Hugo* Belleville boilers. Each has the same number, namely, twenty-eight;

each has the same grate area, *i. e.*, 154 m., 1,660 feet; but whereas the *Leon Gambetta* had 55,680 square feet of heating surface, the *Victor Hugo* has 55,830 square feet. They are,



FRENCH CRUISER "VICTOR HUGO."

therefore, practically identical in every feature, though the later cruiser has a slight advantage in heating surface.

The principal particulars of these vessels are set out in the following list:

Length between perpendiculars	148.35 m.=486 ft. 9 in.
Breadth, extreme	21.40 m.=70 ft 3 in.
Displacement during trials	12,606 tons.
Ordinary draught	8.10 m.=26 ft. 7 in.
Draught, fully loaded	8.50 m.=27 ft. 11 in.
Total bunkers capacity	2,100 tons.
Contract speed for three hours	22 knot.
Armor:	
Fore, depth above water-line	5.20 m.=17 ft. 1 in.
below water-line	1.40 m.=4 ft. 7 in.
Midships, depth above water-line	2.30 m.=7 ft. 7 in.
below water line	1.40 m.=4 ft. 7 in.
Aft, depth above water-line	2.30 m.=7 ft. 7 in.
below water-line	3 ft. 9 in.
Thickness, fore	90 mm.=3½ in.; 70 mm.=3 in., and 56 mm.=2 in.
midship	170 mm.=7 in.; and 125 mm.=5 in.
aft	90 mm.=3½ in.; and 70 mm.=3 in;
Plates	20 mm.=¾ in.
Armor, athwartships bulkhead, fore and aft	120 mm.=4.8 in.

Lower armor deck, thickness, flat		40 mm.=1.8 in.
	sloping	56 mm.=2.2 in.
Upper armor deck, thickness, flat		20 mm.=0.8 in.
	sloping	34 mm.=1.3 in.
Engines, three in number—	*Leon Gambetta.*	*Victor Hugo.*
Boilers	28 Niclausse.	28 Belleville.
Surface of grate	154 m.^{e} =1,660 ft.^{e}	194 m.^{e} =1,660 sq. ft.
Heating surface	5,173 m.^{e} =55,680 ft.^{e}	5,186 m.^{e} =55,830 sq. ft.
Number of boiler rooms	8	8
Number of funnels	4 of 21 m.=69 ft.	4 of 21 m. 69 ft.
Coal, normal bunkers	1,400 tons.	1,400 tons.

With the full bunker capacity of 2,100 tons the active radius at 10 knots is 12,000 miles, and at full speed 1,500 miles. The armament in the turrets consists of four quick-firing guns of 194 mm., (7.6 inches), and twelve quick-firing guns of 164 mm., (6½ inches); in casemates, of four quick-firing guns of 164 mm., (6½ inches); besides twenty-two quick-firing guns of 47 mm., (three-pounders). There are two torpedo tubes, each of 450 mm., (18 inches). The turret armor has a thickness of 200 mm., (8 inches) for the 7.6-inch guns and 140 mm., (5½ inches), for the 6½-inch guns. The casemates have a thickness in the outside plating of 120 mm., (5 inches); in the top plate of 22 mm., (1 inch); and in floor plate of 30 mm., (1¼ inches). The fore-and-aft fire consists of two guns of 194 mm. and eight guns of 164 mm.; and the broadside fire of four guns of 194 mm. and eight of 164 mm.

The following table gives a comparison of the trials of the *Leon Gambetta* and *Victor Hugo:*

TRIALS.

Number.	Full speed, 3 hours.			Ordinary speed, 24 hours.			Consumption trial, 6 hours.		
	Contract.	*Leon Gambetta.*	*Victor Hugo.*	Contract.	*Leon Gambetta.*	*Victor Hugo.*	Contract.	*Leon Gambetta.*	*Victor Hugo.*
Number of boilers at work	All	28	28	All	28	28	...	4	12
Grate surface m^{2}	...	154	154	...	154	154	...	2,262	12
Mean I.H.P.	27,500	29,008	28,436	16,000	16,911	...	...	2,500	8,948
Consumption per IHP., kilos.	...	29,008	0,777	0,850	0,761	0,658	0,850	0,739	0,651
m^{2} of grate,									
kilos	...	29,008	143	...	83	77	...	81	76
Mean speed, knots	22	23.06	22½	...	20.37	19	...	10.30	...
Maximum of speed	...	24 02	23.10	...	...	...	...	...	...
I.H.P.	...	30,500	29,048	...	...	...	...	...	...

—"The Engineer."

Launch of the French Battleship Vérité.—The Société Anonyme des Chantiers et Ateliers de la Gironde, La Bastide-Bordeaux, launched in the presence of the Minister of Marine on Tuesday, May 28th, at 6·30 A. M., from their yard, situated on the right bank of the River Garonne, the battleship *Vérité*, the principal dimensions of which are:

Length between perpendiculars 133.8 m., feet	439
Main breadth, 24.25 m., feet and inches	79–06
Depth amidships, 12.55 m., feet	41
Draught, forward, 7.98 m., feet and inches	26–02
amidships, 8.20 m., feet and inches	26–11
aft, 8.42 m., feet and inches	27–07
Area of amidship section, 179.75 sq. m., square feet	1,932
Displacement, tons	14,870

The sister ships of the *Vérité* are the *Démocratie*, in course of completion at Brest dockyard; the *Justice*, ordered from the Forges et Chantiers de la Méditerranée, La Seyne; and the *Liberté*, in course of completion with the Chantier et Ateliers de St. Nazaire. The four ships were included in the French Navy programme for 1900, and were put down in 1902, but their construction was suspended during several months, under the Pelletan Ministry, with a view to cut down expenses, and to devote all available funds and all the industrial facilities of the nation to the construction of smaller craft and submarine boats. The submarine craze is, therefore, largely responsible for the time taken in placing the four above-mentioned units on the list of ships available for active service.

The interesting feature in the operation which took place on May 28th at Bordeaux is the fact that the *Vérité* was launched 90 per cent. completed, with her engines and boilers mounted ready, her side and deck armor and her turrets fitted and in place. In one month from now she is to be inspected by the Admiralty Commission; she will then proceed to Brest under steam for her official trial runs. Her launching weight is 12,500 tons; this, we believe, is the heaviest weight

ever launched. In this the hull and armor enter for about 5,000 tons each, and the engines and boilers for about 1,000 tons. The time occupied by the vessel while gliding down the launching ways was about 50 seconds.

Owing to the local conditions of the river and to the absence of a dry dock in connection with their yard, the Chantiers de la Gironde have invariably launched their ships almost completely finished. But the *Vérité* is the largest they have built and so launched, and will be the last they will build under similar conditions. They have recently laid the keel of the *Vergniaud*, one of the six 18,350-ton turbine, 22,500 horsepower, 19-knot battleships of 1906 programme, and have in course of construction a dry dock in which to complete this ship and all others they will build in the future. The *Vérité* was built on a slipway served by two 60-ton steam travelers, 26.6 meters (87 feet 3 inches) high and 30.5 meters (100 feet) wide, which ran over the whole length.

Among their contributions to the larger fighting units of the French Navy we may mention the armored cruiser *Kléber*, 7,700 tons; the cruiser *Protet*, 4,114 tons; the torpedo-boat-carrying cruiser *Foudre*, 6,090 tons; the cruisers *Infernet*, *Chanzy*, and a large number of torpedo boats and ships for the merchant service.

The engines of the *Vérité*, built by Messrs. Schneider & Co. at their Creusot Works, are vertical, three-cylinder, compound, three in number, each driving a propeller. They are mounted in three compartments, separated from each other by longitudinal watertight bulkheads. They develop, including the accessory engines and feed pumps, a total of 18,000 horsepower at 110 revolutions. Steam distribution is by Stephenson link motion and cylindrical valves:

Diameter of high-pressure cylinder, 0.860 m., inches	33 13/16
intermediate cylinder, 1.240 m., inches	49
low-pressure cylinder, 1.920 m., inches	76
Stroke, 1.150 m., inches	45¼

The boilers are of the Belleville type, and are twenty-two in number, their principal characteristics being:

Pressure, 21 atmospheres, pounds per square inch	308
Number of stokeholds	5
Total grate area, 127 sq. m., square feet	1,365
wetted heating surface, 3,600 sq. m., square feet	38,700

The three propellers are four-bladed; the central and starboard propeller are left-handed, and the port one is right-handed. The central propeller is 4.8 meters (15 feet 9 inches) in diameter, and the two lateral ones 5 meters (16 feet 5 inches). The contract speed for the ship is 18 knots.

Her armament is to consist of four 305-millimeter (12-inch) guns, mounted in two revolving armored turrets on the center line of the ship, one forward and one aft; ten 194-millimeter (7.63-inch) guns, six of which are mounted in six armored turrets for broadside firing, and four in four armored "blockhaus;" thirteen 65-millimeter (2.56-inch) quick-firing guns distributed over the 'tween deck; and ten 47-millimeter (1.85-inch) quick-firing guns on the bridges and in the fighting tops. There are, besides, two submarine torpedo-launching tubes for 450-millimeter (17.71-inch) torpedoes. The armament will include, also, six automobile and twenty automatic mechanical torpedoes.

She is protected by belt armor formed of cemented steel plates 280 millimeters (11 inches) thick at the center, tapering down to 180 millimeters (7 inches) at both ends of the ship; also by an upper armored deck of special steel plates 54 millimeters (2⅛ inches) thick, and by a lower armored deck, the horizontal plates of which are 51 millimeters (2 inches) thick, and those on the slant 70 millimeters (2¾ inches). The armor of the rotating turrets for the 305-millimeter (12-inch) guns is 280 millimeters (11 inches) thick, formed of cemented steel plates; the 194-millimeter (7.63-inch) guns are protected by plates of the same quality, 200 millimeters (7.87 inches) in thickness; the "blockhaus" plates are 300 millimeters (11.81 inches) thick.

GERMANY.

A New Turbine-propelled Torpedo Boat known as *G 137*, built at Kiel, developed a speed of 32 knots an hour on her trial trip on July 30 in Eckernfoerde Bay.

Danzig.—This small German cruiser on her trials made 22.96 knots with 12,114 indicated horsepower. She thus fell a trifle short of her designed speed. She is the last of the seven *Bremen* class, and the only one to be under 23 knots on trial. It is interesting to note that with most of these German cruisers those built by contract beat those constructed in the Imperial Dockyards.

The trial speeds obtained by this particular *Bremen* class were as follows:

	Speed, knots.	Builder.
Bremen	23.2	Weser, Bremen.
Hamburg	23.1	Vulcan, Stettin.
Berlin	23.2	Danzig Dockyard.
Lubeck (turbines)	23.5	Vulcan, Stettin.
München	23.4	Weser, Bremen.
Leipsic	23.0	Weser, Bremen.
Danzig	22.96	Danzig Dockyard.

Ersatz Baden.—The contract for the third German *Dreadnought*, the *Ersatz Baden*, has been secured by the Krupp Germania yard at Kiel. The displacement of these ships is now stated to be 19,000 tons. The armament is sixteen 11-inch guns of 50 calibers each. Speed, about 19 knots.

Emden is to be made into a first-class torpedo station for the German Navy. It is already a secondary station, but in future it will be the chief North Sea torpedo base. Cuxhaven is being made into a Rossyth; or, rather, the Germans are doing there what the English might have done in the Humber.

The German Armored Cruiser "F" is to be built at the Weser Yard, Bremen. She is of 19,200 tons displacement; armament, twelve 11-inch; speed, 25 knots; horsepower, 50,000. Turbine machinery.

German Torpedo Boats.—A series of torpedo boats, *G 132*

to *G 137*, is now completed. These torpedo boats have a displacement of 440 tons each and steam at the rate of 27 knots per hour; one of them even exceeded 28 knots per hour on her trial trips. A new series of German torpedo boats is contemplated with a prescribed speed of 30 knots per hour.

Schlesien and Schlesweig-Holstein.—With the launch of these two ships there will be twenty-four German battleships afloat, having each a displacement of from 10,000 to 13,200 tons. A new period in naval construction will now begin, in which the displacement will be increased to 18,000 tons, and the armament will be completely altered. This is expected to increase the cost of building each ship from £1,212,500 to £1,825,000. The twenty-four battleships just mentioned constitute the total output in battleships during the present Emperor's reign. Of the number, eighteen have been built in private yards, and only six in Government yards.

THE ITALIAN NAVY.

In 1898 Messrs. C. & T. T. Pattison, of Naples, were commissioned by the Italian Government to build a number of 30-knot destroyers. The design and working drawings of the hulls and machinery were supplied by Messrs. John I. Thornycroft & Co., of Chiswick and Southampton, the boats being of a similar type to those built by them for the British and Japanese navies. The dimensions of this class are as follows: Length, 210 feet; beam, 19 feet 6 inches; draught, 7 feet 6 inches. The engines are of the four-cylinder, triple-expansion, Thornycroft, inclined, condensing type each set having four cylinders as follows: High-pressure cylinder is 22 inches in diameter, intermediate cylinder 29 inches in diameter, and two low-pressure cylinders each 30 inches diameter. The stroke is 18 inches. The contract is for 6,000 I.H.P. and the designed boiler pressure is 220 pounds.

The armament consists of one 12-pounder quick-firing gun and five 6-pounder quick-firing guns, and there are two torpedo tubes.

All the eight destroyers of this class have been launched and have fulfilled most satisfactorily the guaranteed conditions, each boat attaining a speed of at least 30 knots. It is interesting to note that they were the first destroyers to be built in Italy, and they reflect great credit on the builders, as the results obtained with them were as good as those obtained by Messrs. Thornycroft in England with boats of similar design.

Messrs. Pattison also obtained orders for four first-class torpedo boats from designs supplied by Messrs. Thornycroft, which resembled boats recently built at Chiswick for the British Government, except that twin screws were fitted instead of single screws. These boats also have gone through their trials with great success.

The following table shows the mean speeds obtained on the three-hours' official trial of six of the torpedo boats:

Pegaso, ...	25.46 knots.
Perseo, ...	25.72 knots.
Procione, ...	26.02 knots.
Pallade, ...	26.63 knots.
Cigno, ...	26.40 knots.
Cassiopea, ...	25.40 knots.

The contract speed was 25 knots.

It will be seen that this was well exceeded in every case, the *Pallade*, for example, making 26.63 knots.

Subsequently to the placing of these orders the Italian Government called for tenders from Italian builders for destroyers and torpedo boats which were to be of Thornycroft type. Messrs. Ansaldo Armstrong & Co., of Genoa, and Messrs. Odero fu Alesso, of Sestri Ponente, became licensees of Messrs. Thornycroft in Italy, and orders were placed with the former firm for four destroyers and with the latter for six torpedo boats, Messrs. Pattison also getting six additional torpedo boats.

In the design of the new destroyers Messrs. John I. Thornycroft & Co., Ltd., at the desire of the Italian Government,

arranged for complete coal protection of the engines as well as the boilers of both types of boat. In the destroyers the engines are echeloned and each engine is in a separate watertight compartment.

The leading dimensions are: Length, 211 feet 6 inches; beam, 20 feet 0 inches; draught, 7 feet 6 inches.

The engines are of the same type as those of the *Nembo* class already described, but the cylinders are 21-inch, 28½-inch and two of 30-inch diameter by 18-inch stroke. There are, as in the *Nembo* class, three Thornycroft water-tube boilers.

The armament consists of four 12-pounder quick-firing guns, and there are three torpedo tubes. The full complement of men is fifty-three in both the *Nembo* and *Bersaglieve* classes.

It may here be mentioned that Messrs. Ansaldo Armstrong & Co. are also now building six stock destroyers identical in all respects with those of the *Bersaglieve* class for the Italian Admiralty.

Twenty-four of the thirty Thornycroft water-tube boilers required for these ten destroyers have been or are being built by Messrs. Thornycroft at Chiswick, the remaining six being built, under license, by Messrs. Pattison at Naples. The leading dimensions are as follows: Length, 164 feet 0 inches; beam, 17 feet 4 inches; draught, 7 feet 0 inches.

The engines are of Thornycroft type and consist of two sets of vertical, triple-expansion condensing engines with three cylinders, 17 inches, 23½ inches and 36 inches in diameter, the stroke being 14 inches. They are capable of developing about 3,000 I.H.P. There are two Thornycroft water-tube boilers in separate watertight compartments, which supply steam at 200 pounds pressure. The armament consists of three 3-pounders and there are three torpedo tubes.

Three of these vessels have been tried and have exceeded their contract speed, the *Calliope* having attained a speed of 26.38 knots.

Trial Results of the *Regina Elena* in her 24-hours' trials

at four-fifths power showed 15,200 horsepower—I.H.P. 15,-473=20.33. The *Vittor Emanuele* in her preliminary runs at four-fifths for eight hours made 16,000 I.H.P. and 21 knots.

JAPAN.

The report that the Japanese battleships *Aki* and *Satsuma* have turbine machinery is incorrect. Both are fitted with reciprocating engines. The two new battleships in hand are, however, to have turbines. The report that one of these will be built in England is premature, because preparations for building both have been made in Japan. A sum of £2,500,-000 has, however, been allotted to replace ships lost in the late war, and it is not impossible that, Japanese yards being full, some of this money will find its way to other countries.

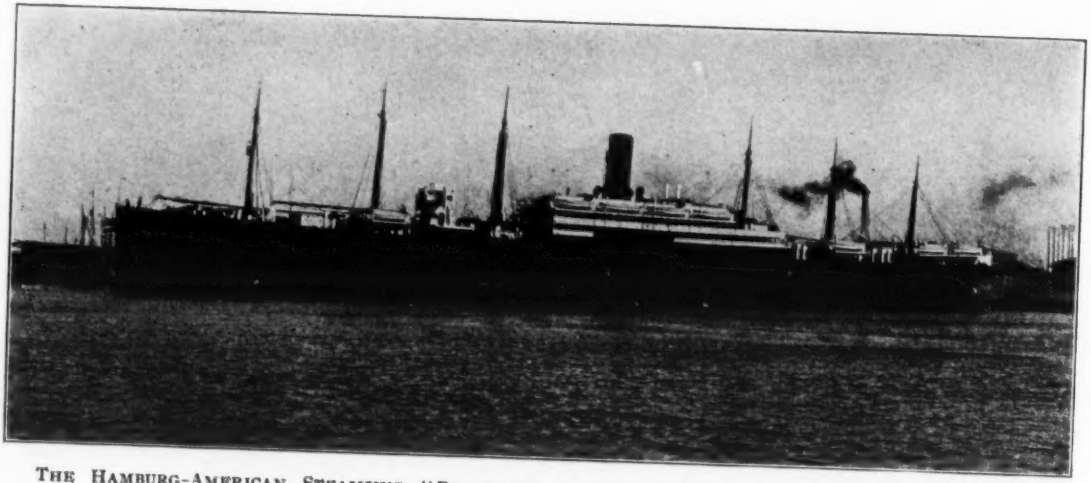
RUSSIA.

Preparations for building a *Dreadnought* have commenced in Russia at Galernii Island Dockyard. This ship will carry ten 12-inch, but be of over 22,000 tons displacement. It is estimated that she will cost well over ten million dollars.

MISCELLANEOUS.

Portugal intends adding to her fleet two destroyers, six torpedo boats and two submarines.

Two Austrian Submarines have been ordered in Germany. They will be of the same general type as "*U*," but larger, as the displacement is about 300 tons. The horsepower will be 300, and the surface speed from 12 to 13 knots.



THE HAMBURG-AMERICAN STEAMSHIP "PRESIDENT LINCOLN;" THE FIRST SIX-MASTER SINCE THE "GREAT EASTERN."

MERCHANT SHIPS.

The New Hamburg-American Liner President Lincoln.—This new steamer, which is a sister of the *President Grant* (soon to appear), reached New York on her maiden trip, after a stormy voyage, early on the morning of June 13, and sailed for Hamburg June 22. The ship covered the distance of 3,047 nautical miles from Cuxhaven to Sandy Hook in 9 days 18 hours and 54 minutes, the average speed being 12.97 knots. The days' runs were 360 nautical miles as a maximum and 241 as a minimum, due to bad weather. The maximum represents a speed of 15 knots and the minimum of 10.04 knots.

These steamers were built by Harland & Wolff, Limited, Belfast, and were intended originally for the Leyland Line. Before completion, however, both vessels were taken over by the Hamburg-American Line. They measure 615 feet in length, with a beam of 68 feet 6 inches, a gross tonnage of 17,540 tons, and a net tonnage of 11,233 tons. The hull is constructed of steel, with a cellular double bottom extending the full length, and watertight bulkheads form a subdivision intended to provide fully for safety. The steamer presents a unique appearance, largely because of her six pole masts fitted with cargo derricks, this being the first six-masted steamer, since the famous *Great Eastern*, to enter the port of New York. The derricks are all of five tons' capacity, and six are located on each mast, with the exception of the fourth and sixth masts, counting from the bow, which have respectively four and three derricks.

On the promenade, bridge awning and upper decks are provided accommodation for 324 first-class passengers, a number of the staterooms being fitted with a single berth each. The

first-class dining saloon is situated on the upper deck amidships, and covers the full breadth of the vessel. Accommodation is provided for 228 diners. This compartment is finished in polished hardwood, with satinwood and lincrusta panels and a parquet floor. A large smoking room is placed aft on the boat deck, the paneling and framing being of carved oak, and the seats and chairs upholstered in maroon leather. The floor is covered with india rubber tiles, while a skylight with a butterfly design is placed over the center of the apartment. The social hall is located at the forward end of the boat deck, and is paneled in carved satinwood, inlaid and upholstered in moquette. An electrically-operated gymnasium and a room for electric baths and massage have been placed on the boat and bridge decks, while a small room devoted to the wireless telegraph equipment enables messages to be sent to a radius of 150 miles, and received from a somewhat greater distance, from more powerful installations.

The second-class accommodation provides for 125 passengers in two, three and four-berth rooms, located aft of the first-class quarters. The dining saloon will seat all of these passengers at once, and is furnished and decorated in white, with medallions in panels; the smoking room and social hall are also tastefully decorated. Third-class passengers are accommodated on the middle and lower decks aft, where provision is made for 1,004. The dining saloon on the upper deck aft will seat 400, and is paneled and decorated in white. A smoking room is provided for third-class passengers. Steerage accommodations are placed on the middle and lower 'tween decks, for 2,320 persons. This makes a total passenger capacity of 3,773. The crew numbers 344, making a total personnel of 4,117 when all berths are filled.

To deal expeditiously with the large cargo which this vessel will carry there are eleven cargo hatches for the seven holds, and steam winches operate in conjunction with the steel-derrick booms, before mentioned, to load and unload the cargo. On the orlop deck are large store rooms for provisions for the passengers, including refrigerating rooms for meats,

etc. The system of ventilation consists of both natural and artificial types, the latter being by means of electric fans. Electric hoists are installed to convey goods from the store rooms to the galleys and pantries, and a number of electrically-operated machines are in use in the culinary department for various purposes. There are sixteen large lifeboats, besides a provision of life rafts carried in four stacks. The ship is fitted with bilge keels, to minimize rolling.

The propelling machinery consists of two sets of quadruple-expansion engines driving twin screws, with a maximum of 8,500 horsepower and a sea speed of 14 knots. The cylinder diameters are respectively 25, 36, 52 and 75 inches, with a common stroke of 54 inches. At full power the revolutions per minute are 86, corresponding with a piston speed of 774 feet per minute. The low-pressure cylinders are fitted with D-slide valves and a balance cylinder, the other cylinders having piston valves, one for each cylinder. The arrangement of cylinders places the high-pressure forward, with the second intermediate next, then the low-pressure, and the first intermediate aft. The high-pressure and low-pressure valves are forward of their respective cylinders, the other valves being aft. These valves are operated by Stephenson double-bar links, actuated by two eccentrics each. The engines are balanced on the Yarrow-Schlick-Tweedy system.

Steam is furnished by six boilers, of which four are double-ended and two single-ended. There are three furnaces in each end of the double-ended boilers and three in each of the single-ended, making a total of thirty furnaces. The aggregate grate surface is 55 square meters (592 square feet); the heating surface is 2,100 square meters (22,600 square feet); the ratio between the two is 38.2 to 1. The boilers are all of a diameter of 14 feet 8 inches, the lengths being 19 feet 10 inches and 10 feet 9 inches respectively. Howden's forced-draft installation is fitted. The steam pressure is 215 pounds per square inch.

The condensers are located in the back columns of the low and first intermediate cylinders, that is, in the after half of

the engines. The propellers have each a diameter of 6 meters (19 feet 8 inches) and a pitch of 6.1 meters (20 feet); this makes a pitch ratio of 1.017. The shafting is carried out through spectacle frames to avoid the use of outside struts. There are eleven sections in each shaft, the two sections of crank shaft having a diameter of 15½ inches. The thrust shaft has seven thrust collars. The line shaft, in seven sections, is provided with two bearings to each section for steadying purposes.

Four sets of direct-connected generators and engines furnish electricity for both lighting and power purposes. Three of these are of 480 ampères capacity each, and the other of 200 ampères; all are operated at 100 volts, the larger types being run at 300 revolutions per minute. The three larger ones have compound engines with cylinders 10 and 17 by 10 inches, and are located just aft of the engines on the main engine-room floor. The fourth unit has a single cylinder, 9 by 8 inches, and is located in a little compartment above the water line, just aft of the engine room, in order to be available in case of emergency, should the engine room be flooded. All were furnished by W. H. Allen, Sons & Company, Bedford. The refrigerating machinery, which includes a compound steam engine operating a single compressor, was furnished by J. & E. Hall, of Dartford. The anchor winch is from John H. Wilson & Company, Limited, Sandhills, Liverpool.—"International Marine Engineering."

YACHTS.

Alexandra.—The new turbine-yacht, the *Alexandra*, built for His Majesty the King, was launched on Thursday, May 30th. This vessel was the subject of competition on the part of a number of the principal yacht-builders in the United Kingdom, who were required to send designs and models to the Admiralty under conditions which precluded any possibility of the author of the design becoming known to the adjudicators. The plans were considered by a Committee, and after long consideration the order for the new yacht was placed with Messrs. J. & A. Inglis, Limited, of Glasgow. This was regarded as well-merited recognition of the undoubted reputation of the firm for their excellence in this department of naval architecture. The result will be looked forward to with considerable interest.

The new yacht is to serve as an auxiliary, rather than as a substitute, for the *Victoria* and *Albert*, built in 1899, but there can be no doubt that, owing largely to the more convenient size, and especially the lighter draught, the new vessel will be preferred for most purposes. The *Victoria* and *Albert* has a length of 380 feet, a beam of 50 feet, and her designed draught was 18 feet, equal to a displacement of 4,700 tons. The new yacht is 275 feet long, of 40 feet beam, and 12 feet 6 inches draught, and she is expected to displace 2,050 tons. She has a bridge-deck 150 feet long, which is carried over the full width of the ship, being supported at the sides on stanchions, so that there is afforded a sheltered promenade. The pavilion covered by this deck contains the reception room, dining room and pantry, with two small boudoirs arranged to afford a clear view ahead and astern as well as on each broadside. Forward under the bridge there is the smoking room, along with the rooms of the commander of the ship, and the surgeon and other officers. On the main deck in the after part of the ship is the suite of royal rooms, with rooms for the secreta-

ries, equerries, ladies-in-waiting, and other attendants. The yacht is also most admirably arranged for the accommodation of guests, officers and crew.

A feature is the adoption of the Parsons turbines for the propulsion of the ship. The turbines are designed to develop 4,500 horsepower, to give a speed of 18¼ knots. The usual three-shaft arrangement—one now universal in the mercantile marine—is adopted, with one high-pressure and two low-pressure turbines, and reversing turbines incorporated in the exhaust casing of each of the low-pressure turbines. The turbines have been designed according to the latest practice in the light of experience obtained. Special attention has been given to accessibility and to convenience in the overhaul of the various parts with a minimum of disturbance of the remainder of the engines. All the bearings are under forced lubrication, with all the latest fittings and sight-connections for the observation of free flow of oil through the bearings. None of the bearings is water jacketed, as in earlier turbines; a special independent oil cooler is provided for cooling the oil. With a view of maintaining a high vacuum the Parsons patent vacuum augmentor arrangements have been fitted in this vessel.

The starting platform and operating valves are on a level with the turbines, so that the engineer has a full view of the whole of the engine room and auxiliaries under his charge. For starting, there is only one simple valve. The maneuvering valves—*i. e.*, one for the low-pressure ahead and the other for the astern—are combined, and operated by one handle. The starting gear is arranged so that these valves are practically horizontal.

NOTES OF THE SOCIETY.

At a special meeting of the Council held on June 25, 1907, the Secretary-Treasurer was authorized to purchase one one-thousand dollar bond of The Washington Railway and Electric Company. This bond has since been obtained.

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